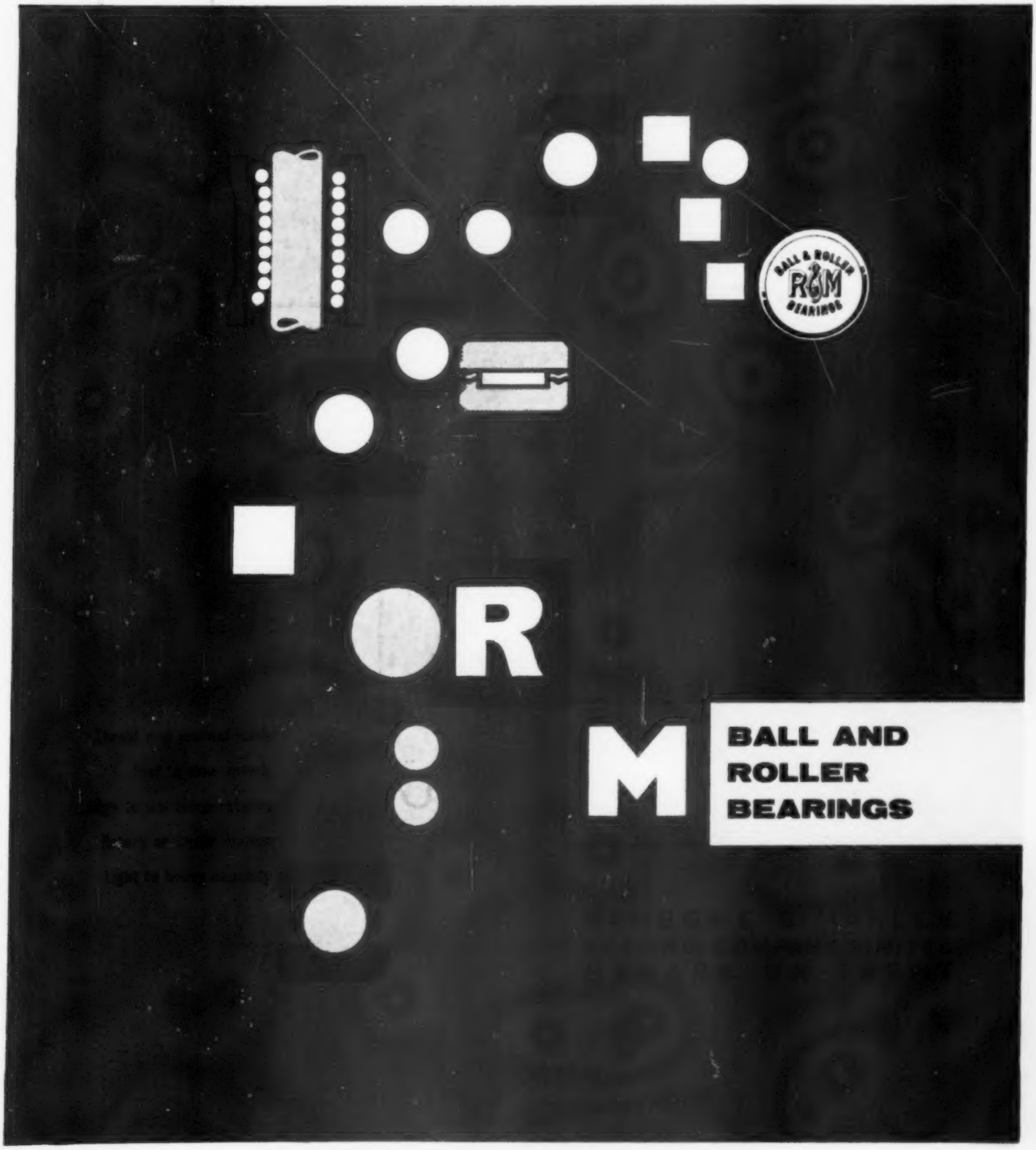
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 50 No. 5 MAY 1960 PRICE: 3s. 6d.

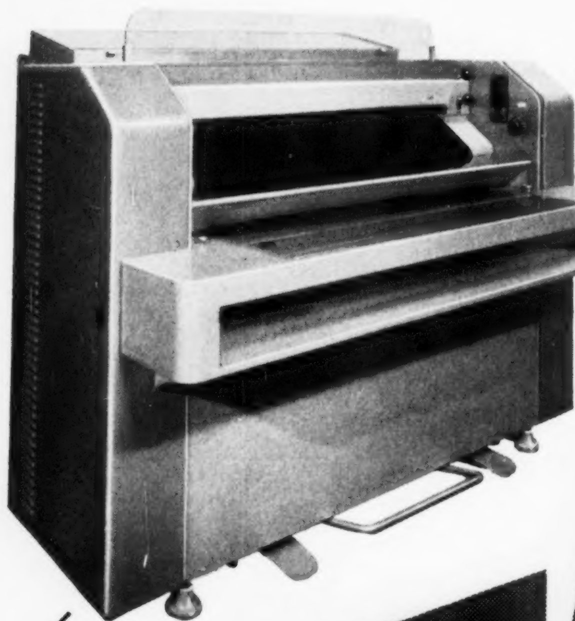


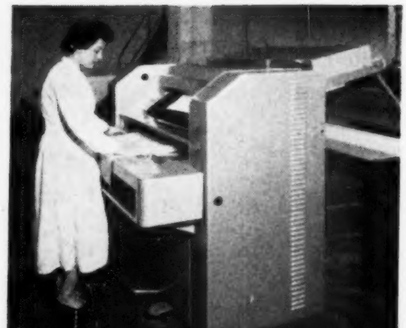
Azoflex Equipment will be demonstrated on
STAND No. B70
INSTRUMENTS, ELECTRONICS & AUTOMATION EXHIBITION
Olympia May 23—28

Rapid, high-quality photoprinting

and no ventilating system required

The Ilford AZOFLEX Model 246 Combine printing and developing machine (formerly known as Model 46/35) is designed for use in the print room of the large drawing office. It does not produce unpleasant fumes and special ventilating systems are thus unnecessary, making it a simple matter to move the machine to a new position at any time.





- Exposure, development and print delivery synchronized for simplicity of operation.
- All controls conveniently located for rapid, effortless adjustment.
- Pneumatic-assisted handling of originals and sensitised material to obviate fatigue.
- Complete design co-ordinated for exceptionally high potential output.
- Excellent mechanical layout giving silent, vibrationless running.
- Comprehensive maintenance service available at nominal cost.

Capacity: rolls and cut sheets up to 42 in. wide.
Printing speed: from 2 ft. to 30 ft. per minute.
Lamp: H.P.M.V. quartz, 3,000 watt.
Dimensions: height, 58 in., width, 72 in., depth (tray extended) 80 in. Weight : approx. 1,400 lb.

Subject to certain conditions, the majority of AZOFLEX photoprinting machines can be hired as an alternative to outright purchase.

ILFORD *Azoflex*

PHOTOPRINTING MACHINES & MATERIALS

Full details from
ILFORD LIMITED, INDUSTRIAL SALES DEPT. AZ18C
ILFORD, ESSEX. TELEPHONE: ILFord 3000

The most powerful surface grinder in its class!

NEW ATLAS COPCO

LSS82 Grinder – a multi-purpose machine

Surface grinding operations with flared cup wheel.

Cutting of a water pipe with high speed cut-off wheel.

Cleaning with wire brush.

High speed cleaning with hub wheel.

This expertly designed, heavy duty surface grinder, with a high-efficiency vane-type air motor, gives you greater power than any other surface grinder in its class. An exceptionally high air flow efficiency has given this model a 35% power increase over its predecessor. Yet for all its power it weighs only 11 lbs.

The new grinder is extremely versatile. Fitted with flared cup wheels, it can be used for cleaning castings and other surface grinding operations; with cut-off wheels attached, it can cut all types of metal, ceramics and stone! Wire brushes can be fixed to clean furnace scale, rust or paint from all large surfaces, and many high-speed cleaning jobs can be done by fitting hub wheels.

DESIGN FEATURES

Highly efficient vane-type air motor with cylinder, rotor and end plates precision worked from special steel alloys to give maximum life and efficiency.

Patented speed governor of a new design which holds speed constant at all loads and gives complete protection from wheel breakage due to excessive speeds.

Safety throttle of entirely new patent construction completely eliminates leakage.

Efficient exhaust silencer, integral with the motor housing, reduces exhaust noise to a comfortable level.

A complete range of compressed air equipment

Atlas Copco manufactures portable and stationary compressors, rock-drilling equipment, loaders, pneumatic tools and paint-spraying equipment. Sold and serviced by companies or agents in ninety countries throughout the world.

Atlas Copco PUTS COMPRESSED AIR TO WORK FOR THE WORLD

Contact your local company or agent, or write to

Atlas Copco AB, Stockholm 1, Sweden, or Atlas Copco (Great Britain) Limited, Maylands Avenue, Hemel Hempstead, Herts.

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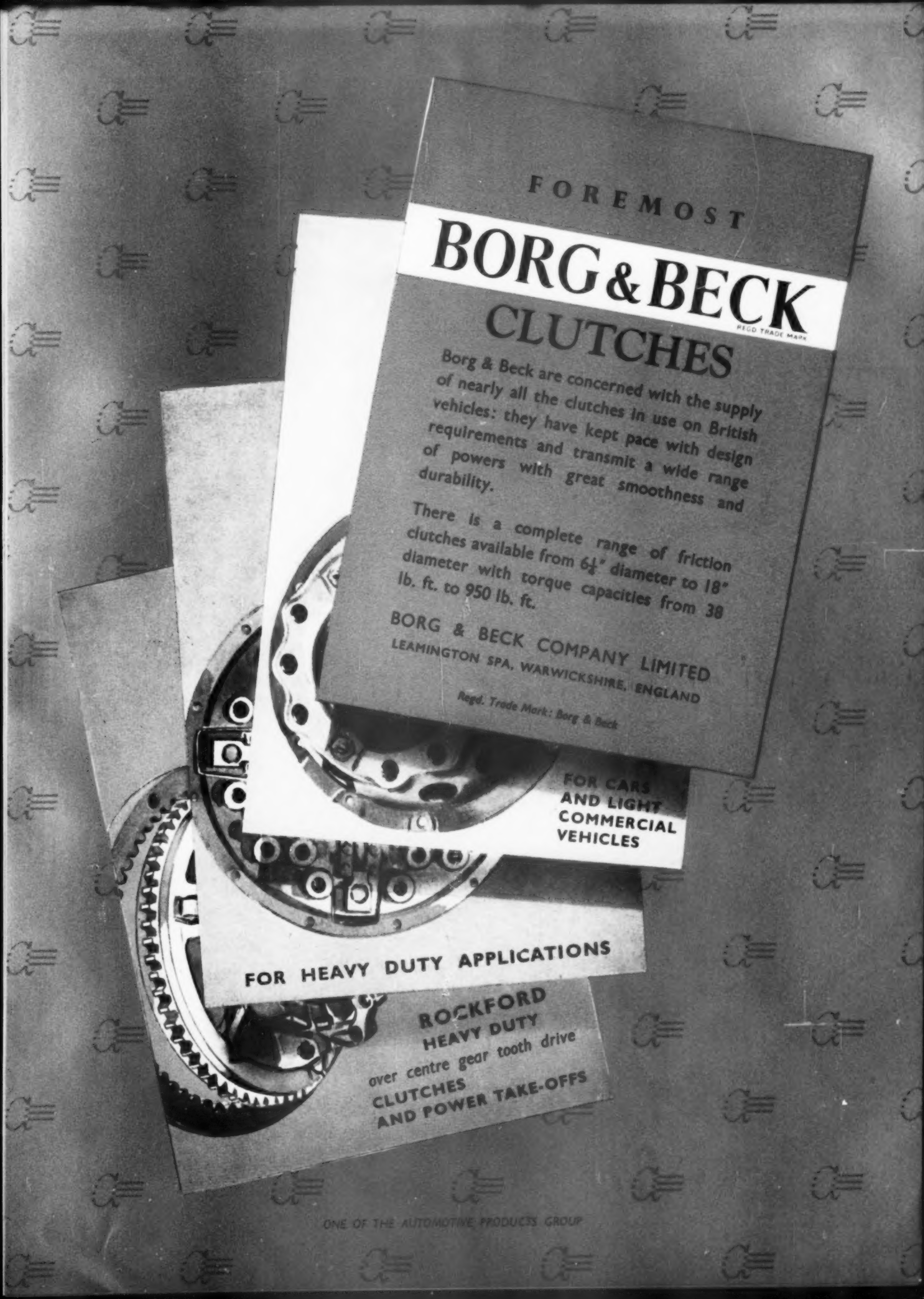
FOREMOST
BORG & BECK
REGD TRADE MARK
CLUTCHES
Borg & Beck are concerned with the supply of nearly all the clutches in use on British vehicles: they have kept pace with design requirements and transmit a wide range of powers with great smoothness and durability.
There is a complete range of friction clutches available from 6¼" diameter to 18" diameter with torque capacities from 38 lb. ft. to 950 lb. ft.
BORG & BECK COMPANY LIMITED
LEAMINGTON SPA, WARWICKSHIRE, ENGLAND
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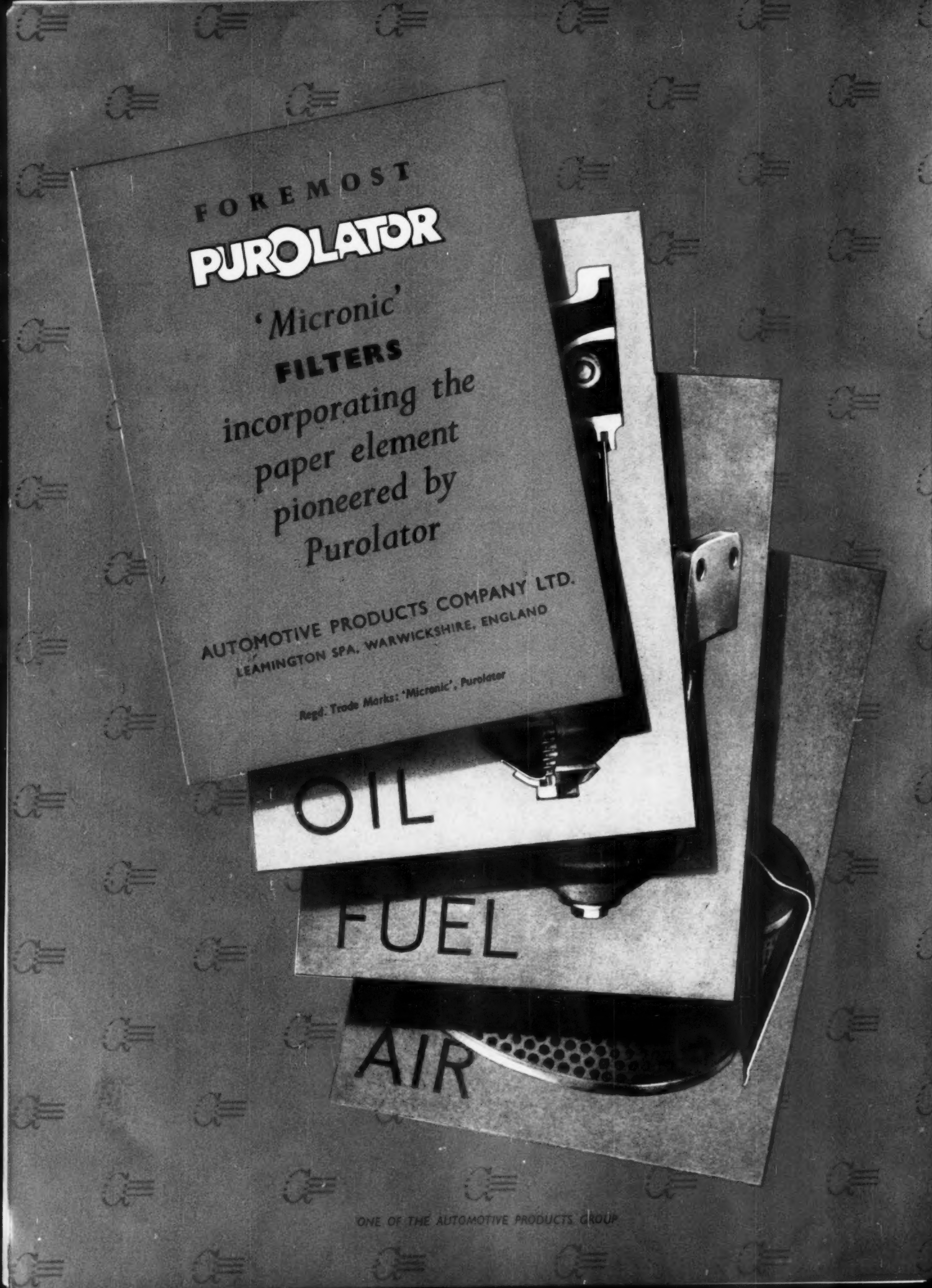
FOR CARS AND LIGHT COMMERCIAL VEHICLES
FOR HEAVY DUTY APPLICATIONS
ROCKFORD
HEAVY DUTY
over centre gear tooth drive
CLUTCHES
AND POWER TAKE-OFFS
ONE OF THE AUTOMOTIVE PRODUCTS GROUP


FOREMOST
PUROLATOR
'Micronic'
FILTERS
incorporating the
paper element
pioneered by
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ONE OF THE AUTOMOTIVE PRODUCTS GROUP

One man and one hundred nuts

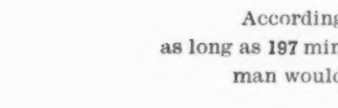
How long should it take him to fix them?

According to accepted engineering practice* it would take a worker as long as **197** minutes to assemble 100 5/16" slotted nuts and split pins. Yet the same man would take a bare 40 minutes to assemble an identical number of Simmonds 5/16" **Nyloc** self-locking nuts.

Common sense says that it pays to use Simmonds self-locking nuts every time when assembly costs run as high as they do today. In fact, with average labour costs and overheads, there are savings of over 42/- on every hundred assemblies—and over £2,000 on every hundred thousand assemblies. Why not call in Simmonds to carry out a completely thorough costing of *your* assembly operations.

Our 16 mm Nyloc colour film is available for showing in your factory.

All times shown are based on "The Handbook of Standard Time Data for Machine Shops" by Haddon & Genger, published by Thames & Hudson Limited, London.



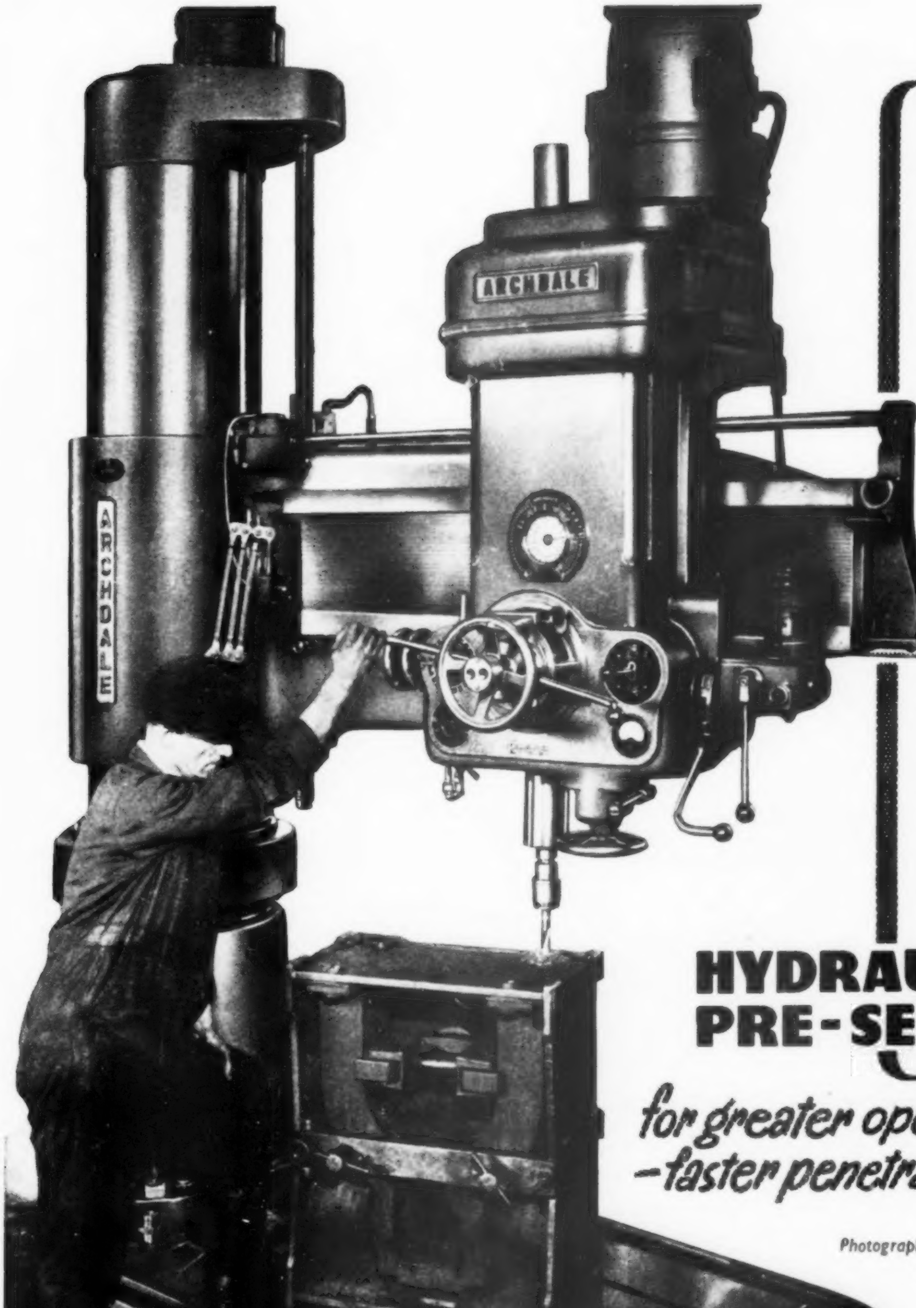
time saved is money saved

SIMMONDS SELF-LOCKING NUTS

SIMMONDS AEROCESSORIES LTD · TREFOREST · PONTYPRIDD · G AMORGAN *A member of the Firth Cleveland Group*

Branches: London, Birmingham, Manchester, Glasgow, Stockholm, Copenhagen, Ballarat, Sydney, Johannesburg, Amsterdam, Milan, New York, Mannheim and Brussels

CRC 43N



ARCHDALE

HYDRAULIC PRE-SELECT

for greater operating speed
—faster penetrating capacity

Photograph by courtesy of
OTIS ELEVATOR CO. LTD.

JAMES ARCHDALE & CO. LTD.

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Works: BLACKPOLE WORKS, WORCESTER. *Tel: Worcester 27081 (6 lines)*
A Member of the Staveley Coal & Iron Co. Limited Group
Sole Agents : ALFRED HERBERT LTD., COVENTRY. Telephone : Coventry 89221

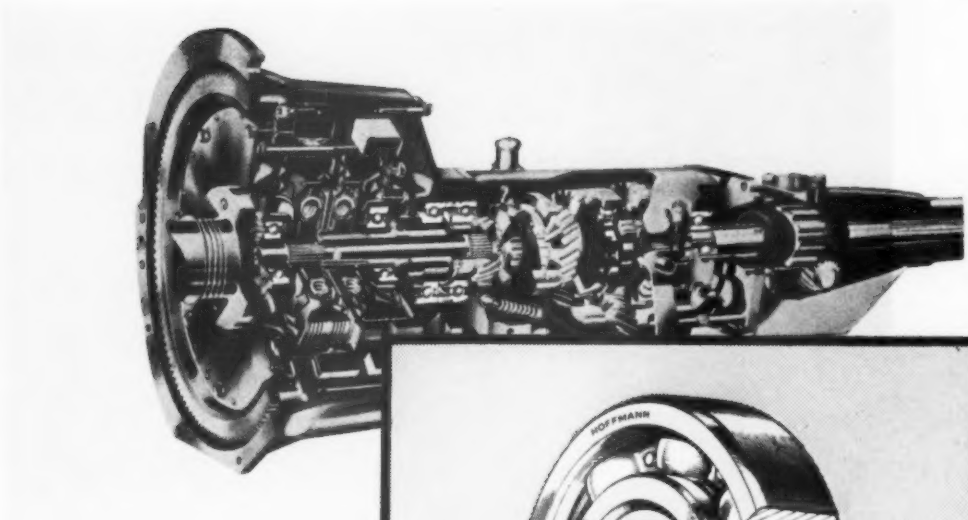
INTERNATIONAL MACHINE TOOL EXHIBITION LONDON 1960 OLYMPIA JUNE 25-JULY 8

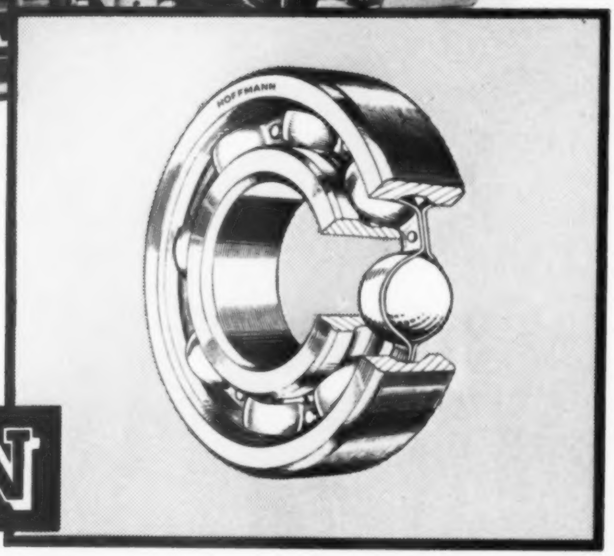
STAND No. 50

SMITHS NEW

FULLY AUTOMATIC

TRANSMISSION FOR THE SMALL CAR





USES

HOFFMANN

BALL BEARINGS

Smiths fully automatic transmission makes use of a layshaft type gearbox but, by the clever application of two magnetic powder couplings and an electro-mechanical control system, gives fully automatic gear changing and elimination of the clutch pedal.

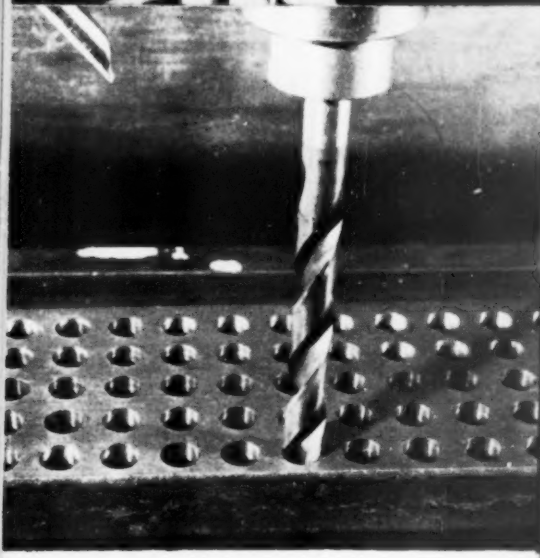
Hoffmann Ball Bearings are fitted in this transmission and contribute much to the smoothness and efficiency of the whole system.

CAN WE HELP YOUR NEW PROJECT? HOFFMANN

THE HOFFMANN MANUFACTURING CO. LTD., CHELMSFORD, ESSEX


HCW
CLATONRITE
SELF SEALING
WEATHERSTRIP
STIKASTRIP
ADHESIVED SPONGE
SEALING STRIP

DO YOU KNOW . . . ?



PRODUCTION

It takes at least thirty people—thirty pairs of skilled hands—to make one DORMER Drill. Every operation is performed by experienced operators with the most up-to-date machinery. Careful heat-treatment and inspection are equally essential in maintaining the consistently high standard of DORMER Tools.

At least one in every ten people employed in the manufacture of DORMER Drills is an Inspector. Throughout production and before each DORMER Tool leaves the works every feature is checked in minute detail.

INSPECTION

PERFORMANCE

With correct use, a DORMER Drill of 1″ diameter, in its lifetime on general purpose work, will remove 2,800 times its own weight in material (over $1\frac{3}{4}$ tons), and drill a total length of hole five hundred yards long!

DORMER

TWIST DRILLS

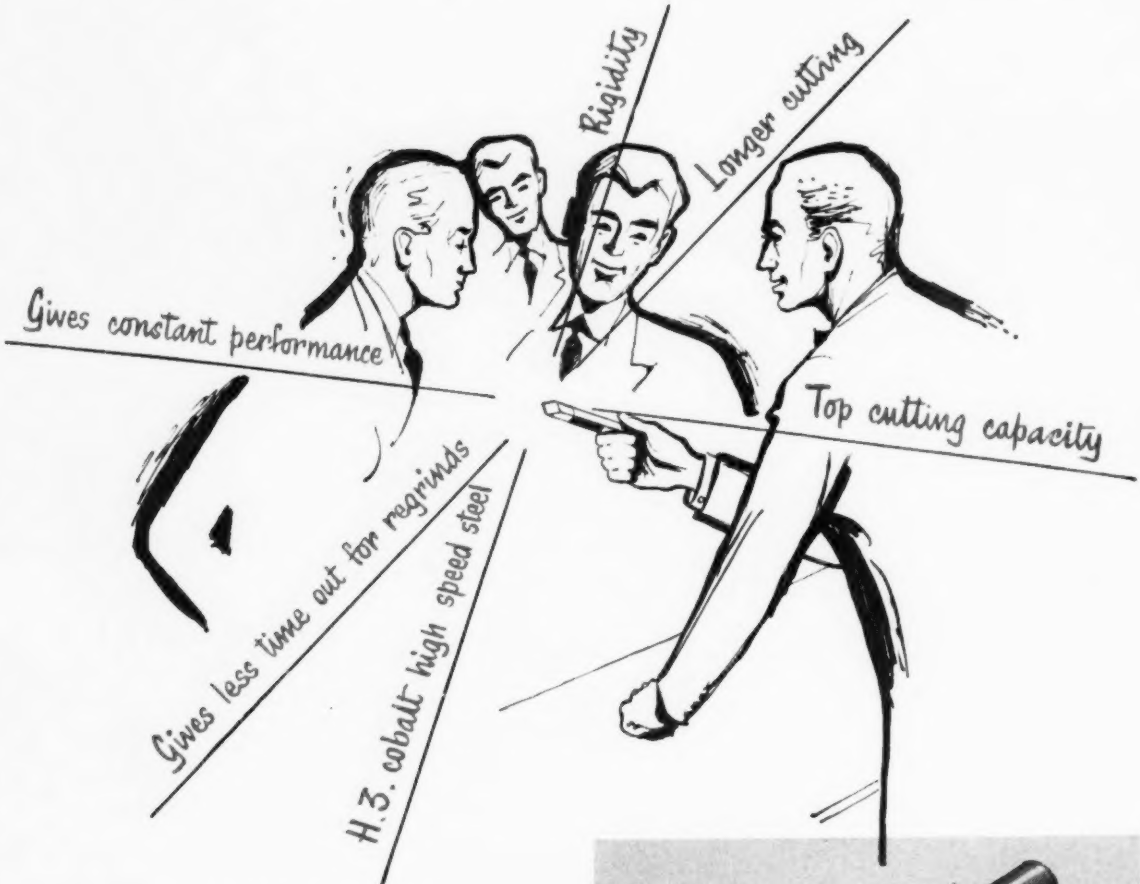
THE FINEST BY ANY STANDARD

THE SHEFFIELD TWIST DRILL AND STEEL COMPANY LIMITED

SHEFFIELD ENGLAND

DORMER TOOLS ARE OBTAINABLE FROM YOUR USUAL ENGINEERS' MERCHANT

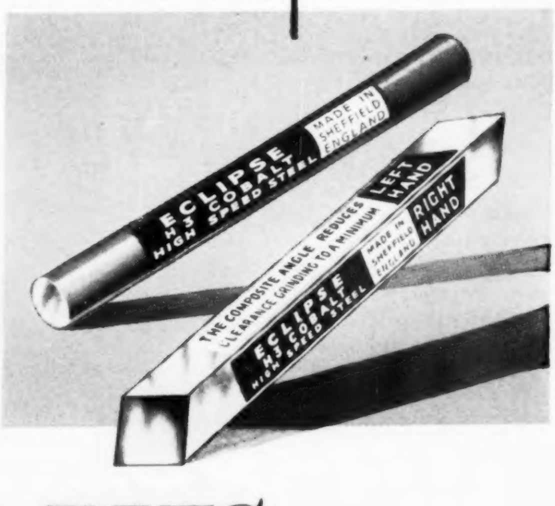
that extra little bit . . .



It is the consistent high quality and the exceptional ability of "Eclipse" tool bits to retain a keen cutting edge that puts that elusive production target within your reach.

A tool bit is only as good as the steel it's made from, and "Eclipse" tool bits are made from the finest H3 Cobalt High Speed Steel. In other words, you can rely upon fast, accurate cutting and know also that time-out for regrinds is reduced to an absolute minimum.

Use "Eclipse" tool bits in "Eclipse" tool bit holders from today; together they provide the finest cutting combination.

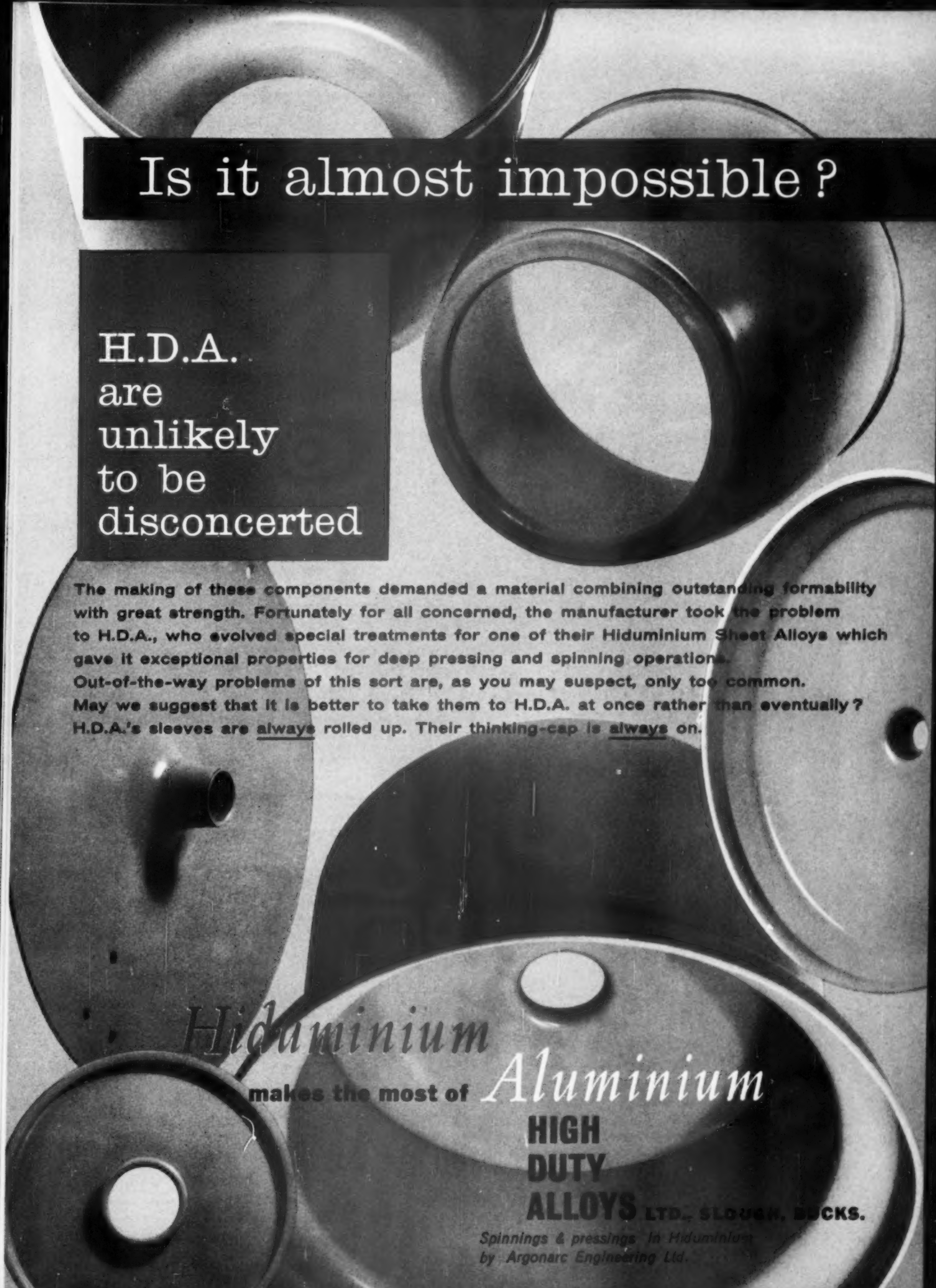


Eclipse

TOOL BITS

'Eclipse' hacksaw blades and other tools are made by James Neill & Co. (Sheffield) Ltd. and are obtainable from all tool distributors

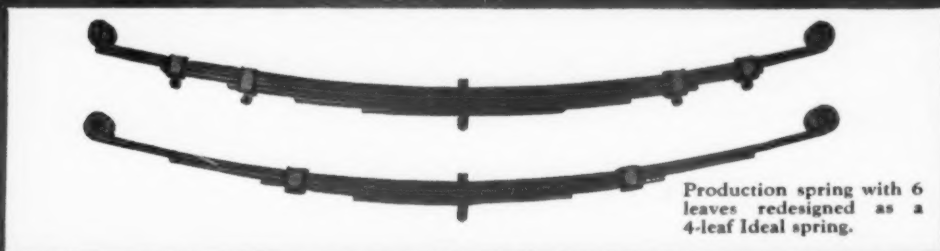
UT15



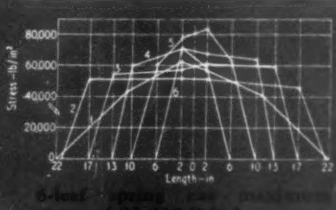
Is it almost impossible?
H.D.A. are unlikely to be disconcerted
The making of these components demanded a material combining outstanding formability with great strength. Fortunately for all concerned, the manufacturer took the problem to H.D.A., who evolved special treatments for one of their Hiduminium Sheet Alloys which gave it exceptional properties for deep pressing and spinning operations.
Out-of-the-way problems of this sort are, as you may suspect, only too common.
May we suggest that it is better to take them to H.D.A. at once rather than eventually?
H.D.A.'s sleeves are always rolled up. Their thinking-cap is always on.
Hiduminium
makes the most of Aluminium
HIGH DUTY ALLOYS LTD., SLOUGH, BUCKS.
Spinnings & pressings in Hiduminium by Argonarc Engineering Ltd.

HOW TO SAVE SPRING WEIGHT WITHOUT INCREASING STRESS

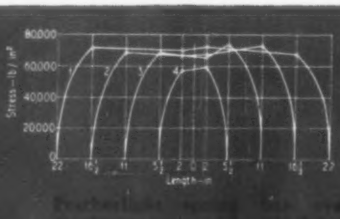
By applying rational design theory we are able to offer springs with scientifically tapered leaves, showing an advantageous stress distribution avoiding localised high stresses. It is possible in TWS 'Featherlight Ideal' springs to achieve weight saving up to 25% or more, with consequent savings in cost—cheaper springs in fact, to say nothing of the advantage of lower unsprung weight and a better ride.
Featherlight Ideals permit designs of tapered main leaves with greater leaf thickness available locally at the eyes.



Production spring with 6 leaves redesigned as a 4-leaf Ideal spring.

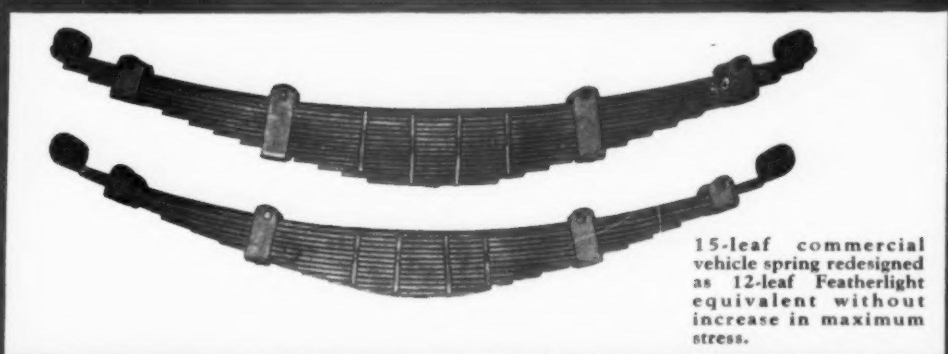


6-leaf spring has maximum stress of 83,000 lb/in².

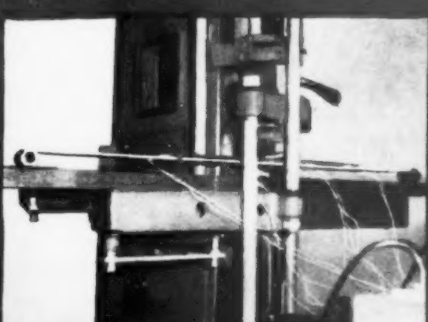


Featherlight spring has even maximum stress of 74,000 lb/in².

Electric strain-gauge measurements demonstrate uneven high stresses of 6-leaf spring and even maximum stress of Featherlight Ideal equivalent.



15-leaf commercial vehicle spring redesigned as 12-leaf Featherlight equivalent without increase in maximum stress.



A tapered Featherlight spring loaded to the test condition, with complete network of multiple electronic strain gauges to measure stresses.

TWS

'featherlight'

IDEAL SPRING

(PATENTS PENDING)

TOLEDO WOODHEAD SPRINGS LIMITED

AYCLIFFE NR. DARLINGTON and SHEFFIELD 3

Dielectric Heating - 2

The ability of dielectric heating to generate heat through the mass of a suitable material provides the following considerable advantages over other heating methods.

1 A body of uniform section and composition is raised in temperature uniformly throughout. Hence there is no waiting for heat to be transferred from an external heat source to the surface of the body and thence to its interior, and this is of particular advantage when the body is thick, and, as is often the case with dielectric materials, has poor heat-conducting properties.

2 The rate of heating of such bodies is, therefore, much faster than by external heating methods.

3 Since there is no external heat source, overheating or burning of the surface of a heat-sensitive material is avoided.

4 High thermal efficiency is achieved.

5 The amount of heat generated in the work is usually predictable and power input and heating time can be under positive control.

6 Production can start immediately after switching on and no current is used, nor heat lost, during periods of shut-down.

7 Vastly increased productivity is obtainable with less labour (usually unskilled), and fewer machines and less floor space are required.

8 Flexibility of layout makes it possible to plan the factory to best advantage, as dielectric heating equipment can usually be inserted directly into the production line.

Dielectric heating: typical application data

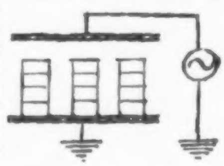
Typical application	Frequency	Radio frequency power
Thermoplastic welding.	20-100 Mc.p.s.	Up to 1 kW
Plastic pre-heating, wood glueing.	10-40 Mc.p.s.	2-30 kW
Plywood manufacture.	2-10 Mc.p.s.	Above 30 kW

Note: 1 Mc.p.s. = 1,000,000 cycles per second.

A few of the industrial applications of dielectric heating are briefly described below.

Preheating Thermosetting Plastics

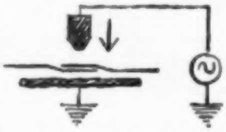
Dielectric heating is the ideal way of preheating moulding powder pellets used in the production of thermosetting plastic mouldings, since these materials are generally poor heat conductors. Properly applied, dielectric preheating promotes faster curing and hence a shorter moulding time, often increasing production ten to fifteen times. There is a marked reduction in tool wear, and thicker sections can be moulded, as the material is plastic when placed in the mould. There is less damage to, and movement of, metal inserts.

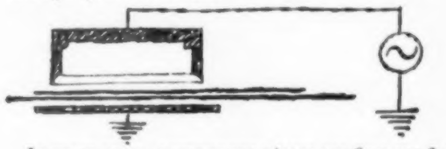


Welding of Thermoplastic Materials

An important and extensive application of dielectric heating in the plastic industry is the welding of thermoplastic sheets in the fabrication of such commonly used articles as raincoats, hoods, handbags, pouches and packaging materials. Dielectric heating is the only method which can usefully be employed since the heating electrodes, and hence the outside sheet surfaces remain cool while the inside surfaces forming the joint are fused, and a perfect weld results.

Two or more thermoplastic sheets are welded under pressure from electrodes suitably shaped to the area of weld required, the current being switched on at the same time as pressure is applied, and off as soon as the weld is completed and the pressure released. Stitching is thus eliminated and a far stronger joint achieved.





In most cases, component pieces are first cut from patterns and preliminary welding carried out to attach any fastening tabs and the like. The pieces are then brought together in a suitable loading frame and the main welding carried out to produce the complete article. In some cases, a suitably profiled electrode can be fitted with a knife edge to cut the sheets immediately outside the weld line, welding and pattern cutting being thereby carried out in the one operation. Dielectric welding can be applied also to very large thermoplastic products such as linings for swimming pools, and cinema screens.

Drying

Drying of materials by dielectric heating has the great advantage that the material tends to dry out from the centre, the reverse of what happens when external heating methods are employed, and the risk of overdrying and overheating of the surface is eliminated. In general ½ to 1 unit of electricity is required to drive off 1 lb. of moisture, depending upon the thermal properties of the material being dried. While the removal of large amounts of water from inexpensive commodities may sometimes be uneconomical, dielectric heating in the production line often leads to a higher overall production efficiency. It is valuable for removing final moisture traces and becomes increasingly economical as the value or heat-sensitivity of the commodity increases.

Rubber

External heating tends to dry and cure the surfaces of a thick latex mass before its centre, but dielectric heating properly applied promotes uniform conditions throughout.

Loaded rubber may not heat uniformly in a dielectric field due to uneven dispersion of its load, but nevertheless rubber preforms loaded up to about 15% are preheated dielectrically to reduce moulding times appreciably, the temperature evening out in the mould to give uniform curing.

Further examples are given in Data Sheet No. 12.

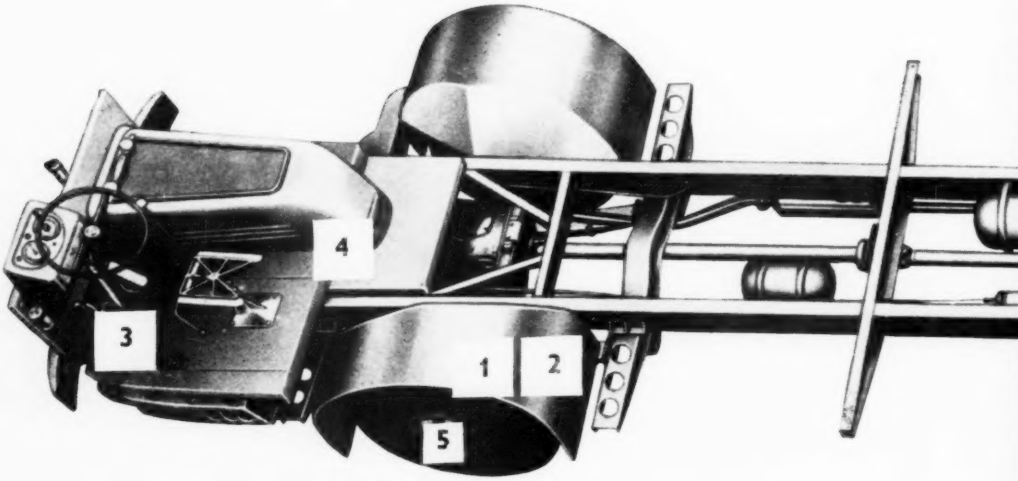
For further information get in touch with your Electricity Board or write direct to the Electrical Development Association, 2 Savoy Hill, London, W.C.2. Telephone: TEMple Bar 9434.

Excellent reference books on electricity and productivity (8/6 each, or 9/- post free) are available — "Induction and Dielectric Heating" is an example.

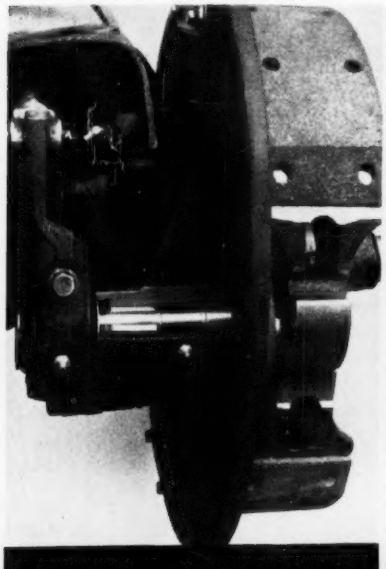
E.D.A. also have available on free loan in the United Kingdom a series of films on the industrial uses of electricity. Ask for a catalogue.

4774

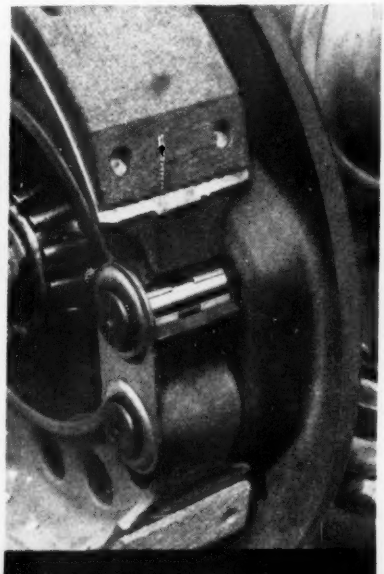
GLACIER DU 'dry'



A.B. Scania-Vabis, Södertälje Sweden are known the world over as makers of one of the finest of all commercial vehicle chassis. They were amongst the first engine builders to fit Glacier 'dry' bearings* and current engine chassis and suspension applications include clutch, brake, and king pin assemblies. Approximately half a million bushes and thrust washers have been supplied to Scania-

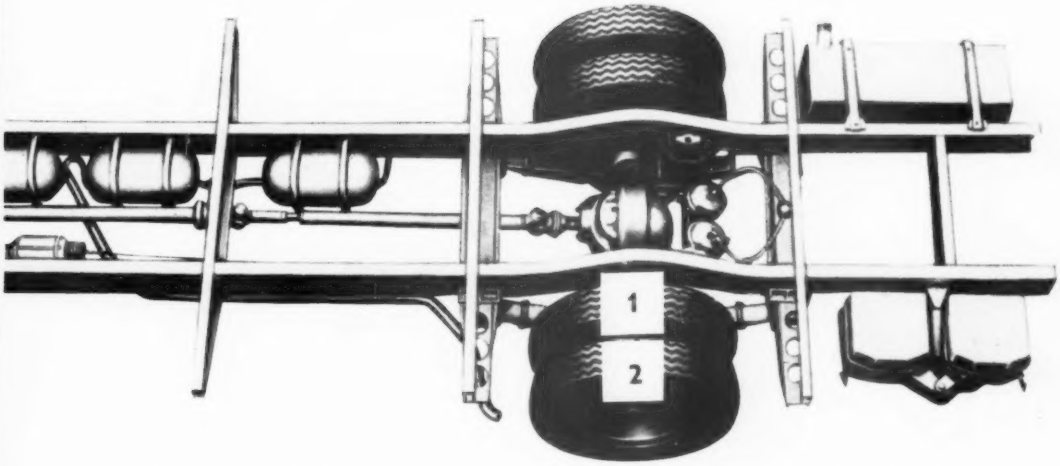


1 Brake Camshaft bearing. Glacier DU bushes in bracket (used in front and rear brakes on all models). Camshaft with chromium plated bearing surfaces.



2 Anchor bolt bearing. Two Glacier DU bushes in brake shoe (used in front and rear brakes on all models). Chromium plated anchor bolt.

3 Clutch pedal bearing. Two Glacier DU bushes in the pedal (used on all models). Chromium plated bearing shaft.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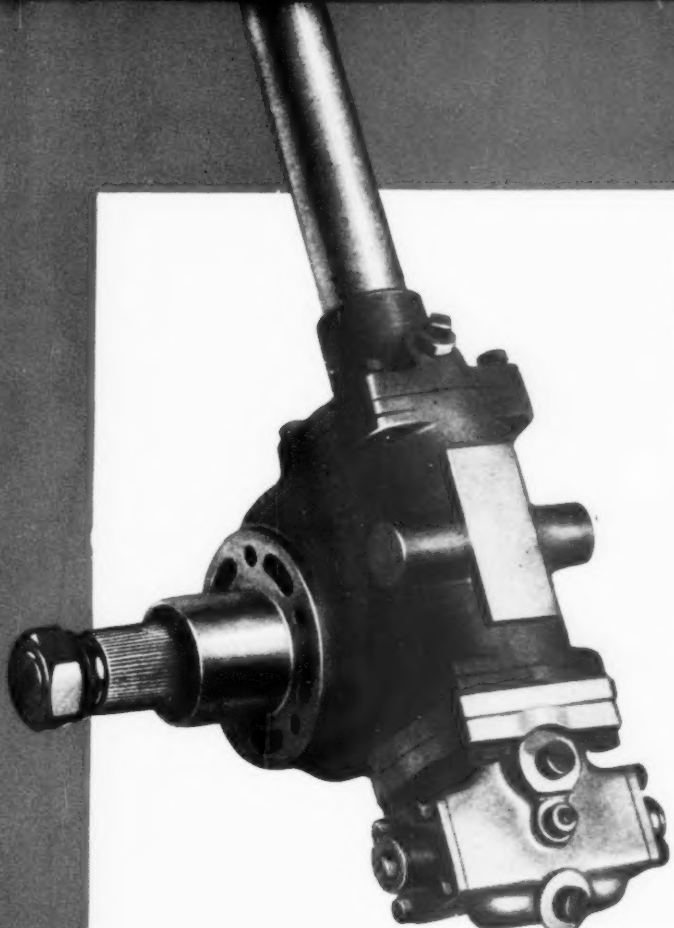


Interchangeable, on the same mounting, with the standard Marles manual gear unit.

THE TYPE 3 'UNIVERSAL' UNIT

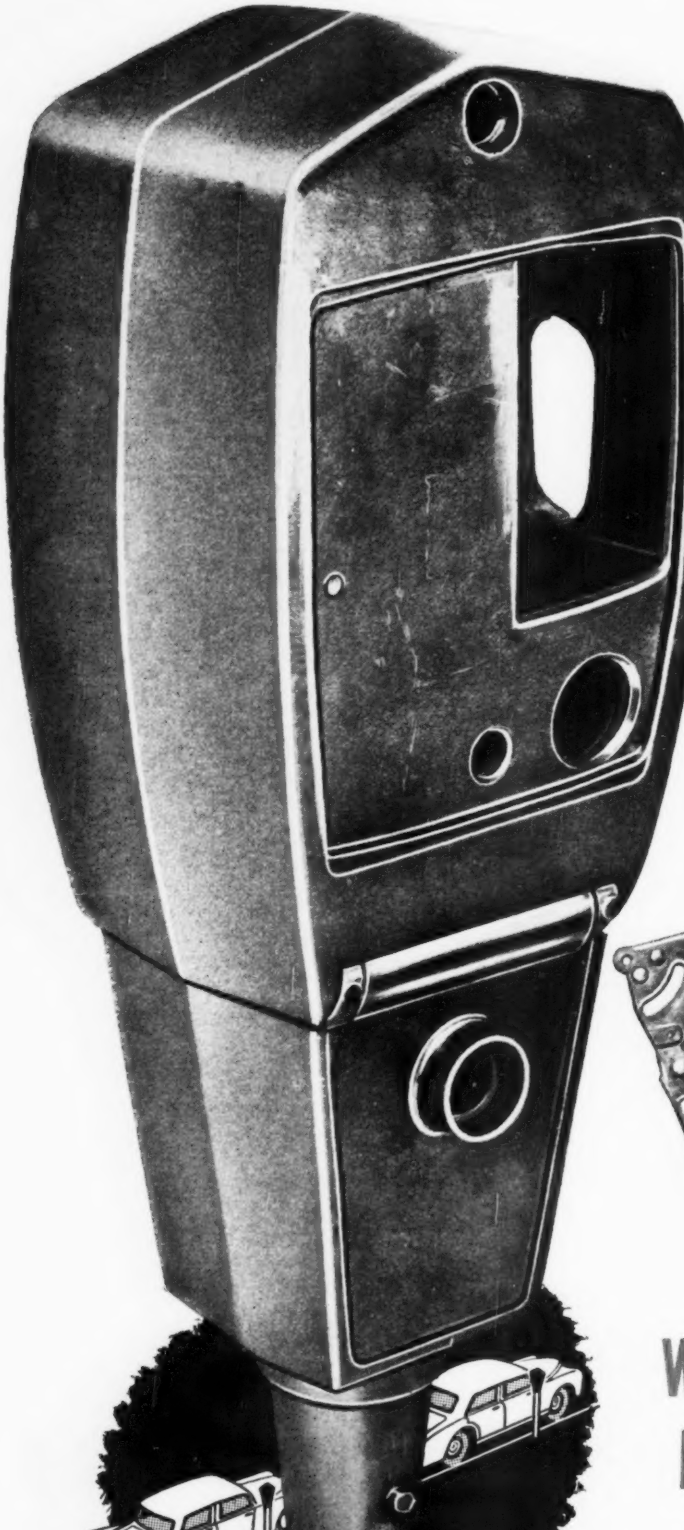
Illustrated above is the Type 3 'Universal' steering gear which incorporates the hydraulic control valves mounted upon our type '861' manual gear. This is for use with a separate power pump and with power cylinders operating on the steering linkage. Further particulars will be sent on request.

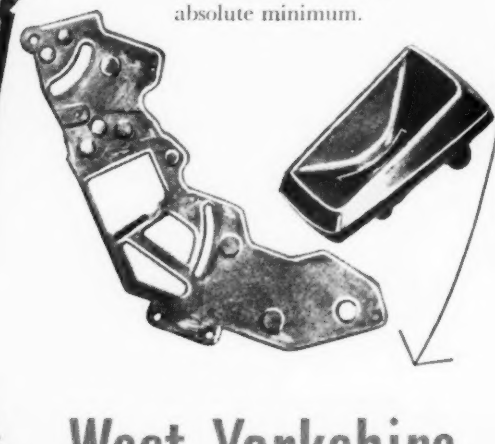
ADAMANT ENGINEERING CO. LTD.,
THE AERODROME, WOODLEY, Near READING.

Sole proprietors of the Marles Steering Company Ltd.

Telephone: Sonning 2351 Telegrams: Adamant, Reading

MARLES





Park your casting problems with us

Having had so many unusual casting jobs 'parked' on us before, it was no surprise when we were asked to produce this parking meter casing and mechanism. The casing is an aluminium gravity die casting whilst the mechanisms are produced by pressure die casting. We also supply sand castings and precision plaster mouldings, so whatever your aluminium casting problem—consult us first. You'll find our modern methods produce not only quality castings in quantity, but they also keep production costs to an absolute minimum.

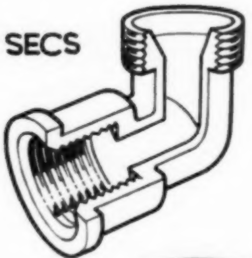
West Yorkshire Foundries Ltd

SAYNER LANE
LEEDS 10
Phone: LEEDS 29466

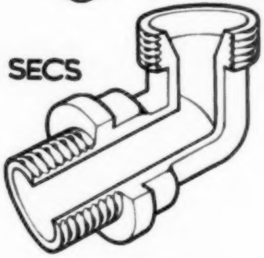
London Office:
HANOVER HOUSE
HANOVER SQ., W.1
Phone: MAY 8561

CAST WELL AND TRUE

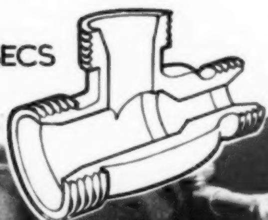
40 SECS



45 SECS



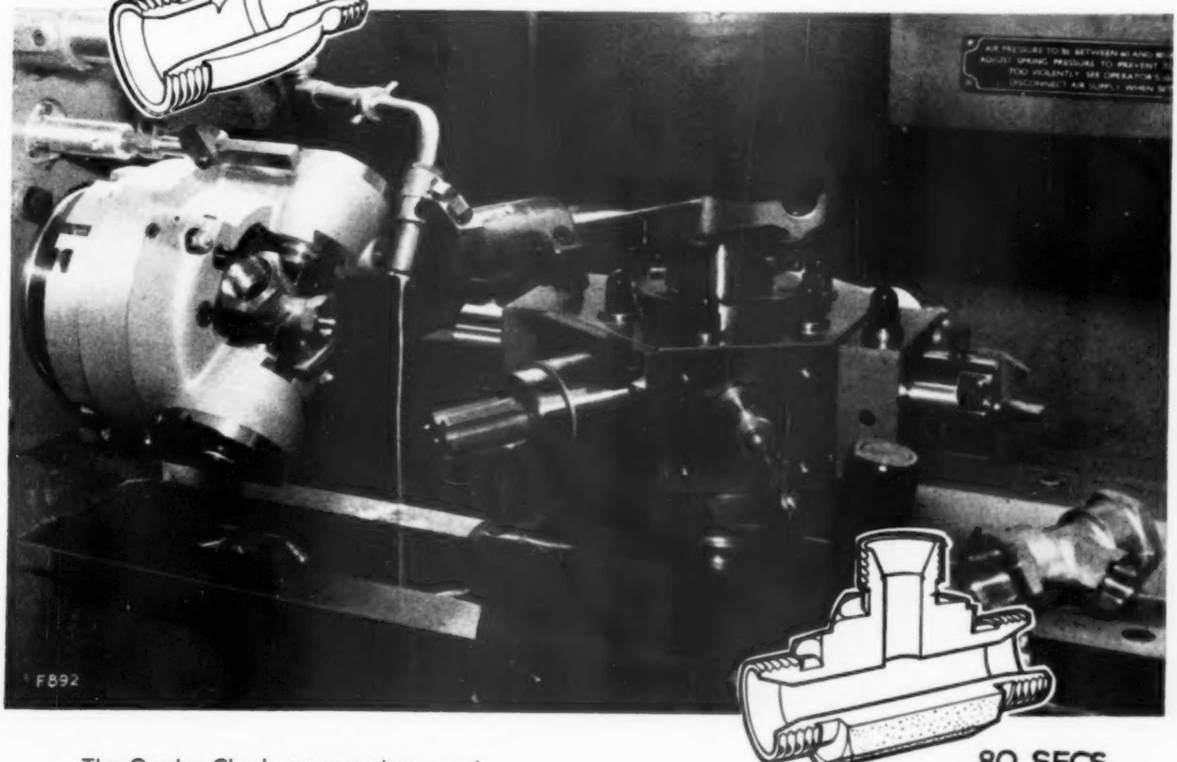
65 SECS



for rapid production of multi-end components on the

HERBERT N°2 FLASHCAP LATHE

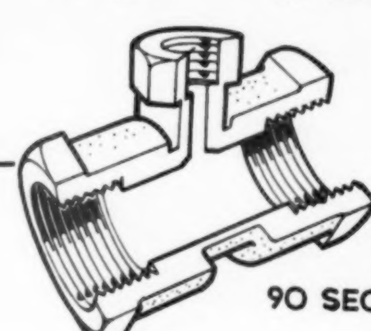
This lathe was specially designed for the high production of non-ferrous threaded components requiring frequent spindle reversals. It can now be fitted with the Herbert Pneumatic-hydraulic Indexing Quaora Chuck, indexed without stopping the spindle, thus providing a simple lathe which considerably reduces machining times in the production of multiple-end components, such as elbows, tees and crosses.



80 SECS

The Quadra Chuck accommodates work which can be indexed in a radius of 2½"; maximum diameter gripped is 1½".

ALFRED HERBERT LTD., COVENTRY



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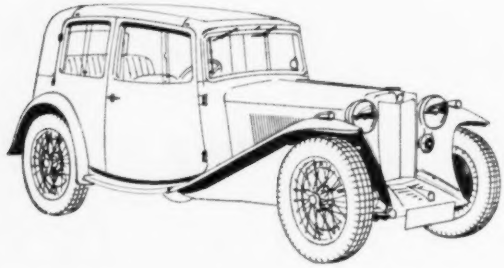

This 1932/3 M.G. Magnette was fitted with a Hardy Spicer propeller shaft. Its brilliant performance soon established it as one of Britain's best-selling, best-looking sports saloons.

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The 1960 M.G. Magnette (Mark 3) sports saloon marks a decisive advance in design and performance on previous models. But the Hardy Spicer propeller shafts and universal joints remain constant features in the transmission. The M.G. Company like other leading car manufacturers, know that the experience of Hardy Spicer, with their consistently advancing methods of research and development, will always meet the demands imposed by increasing strain on transmission equipment.

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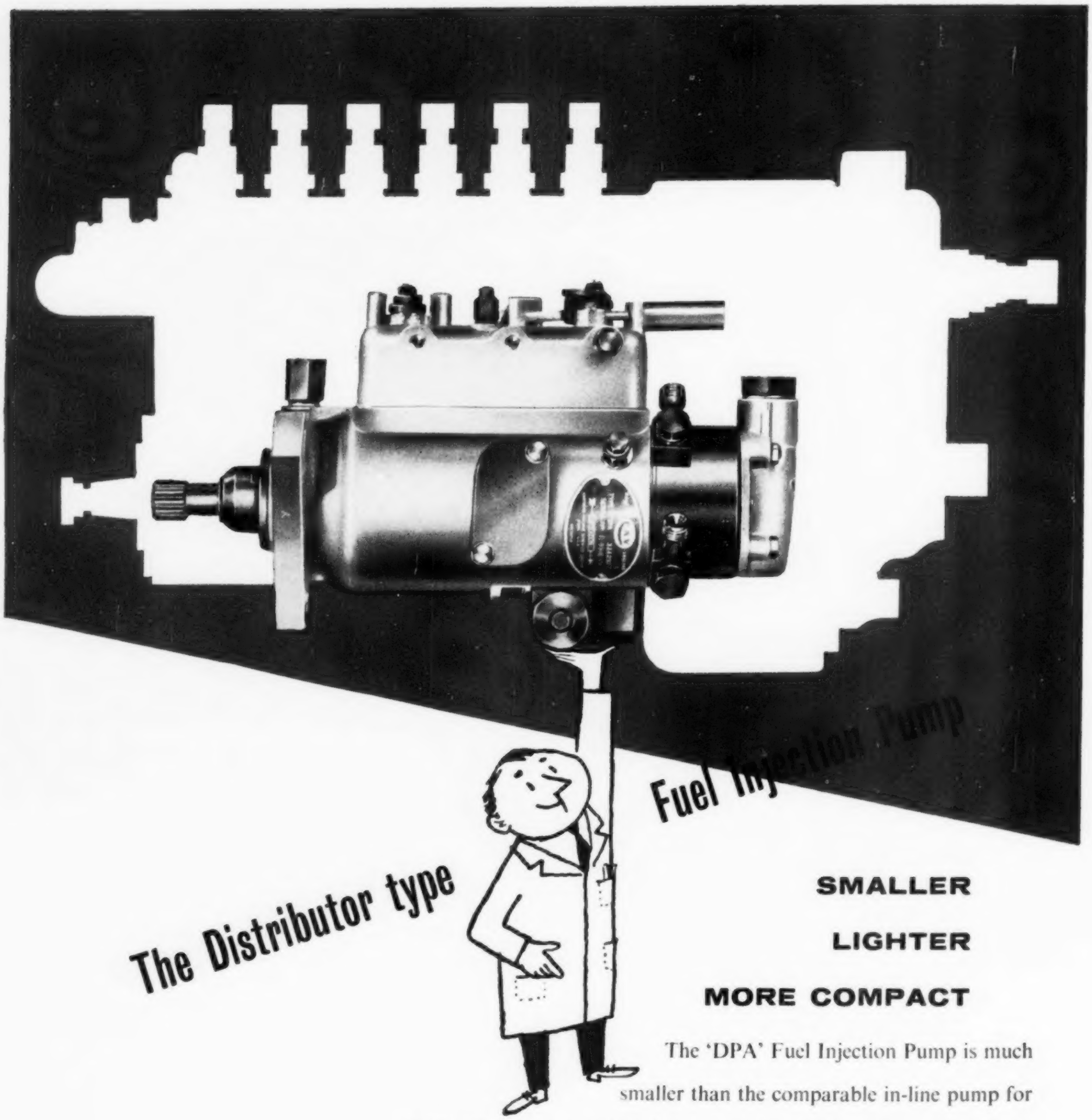

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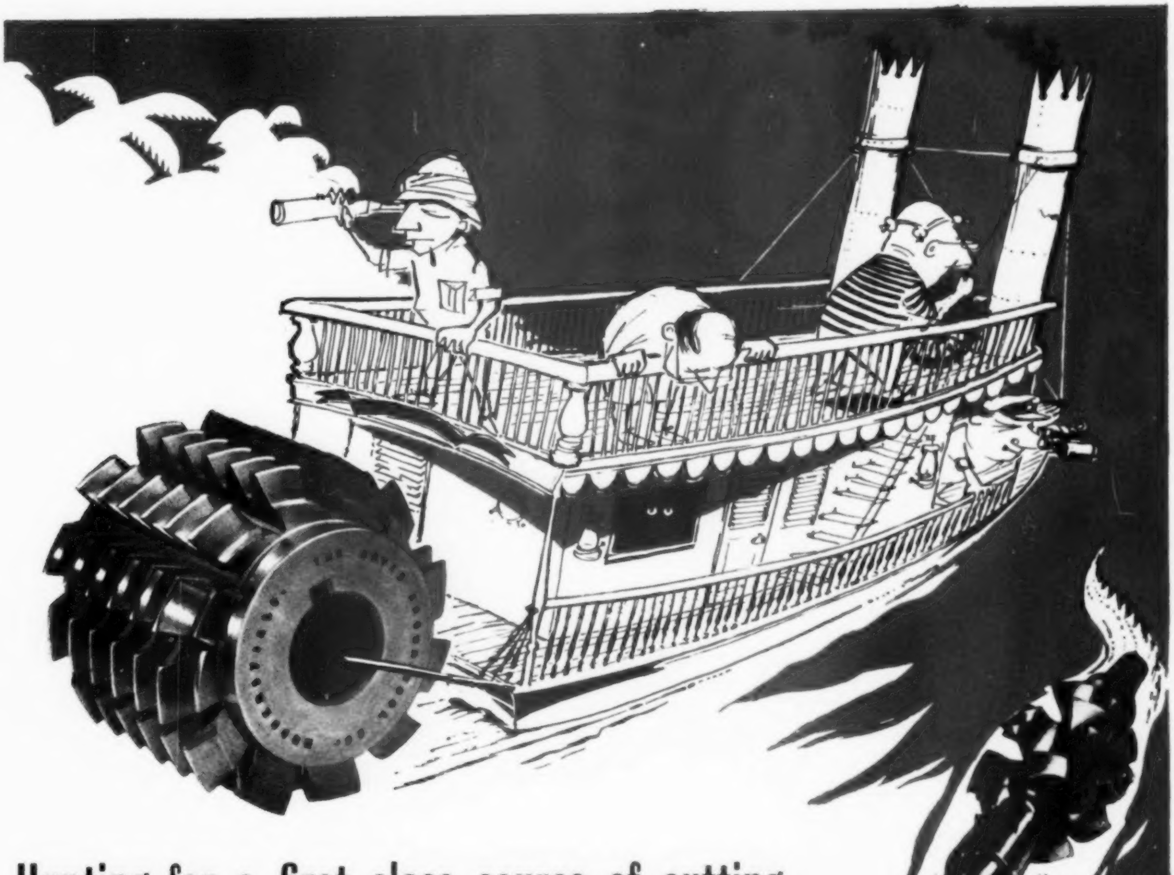

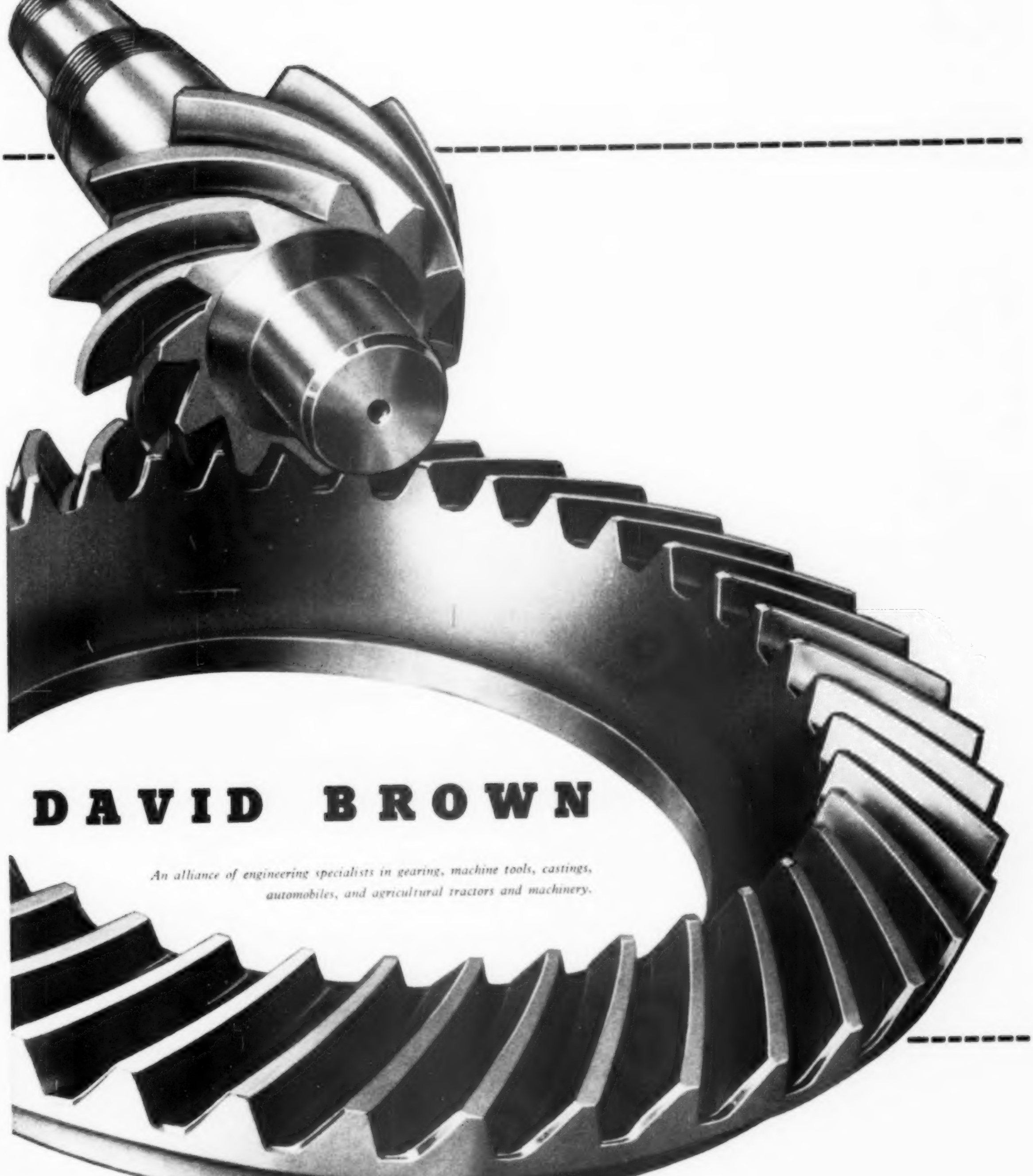
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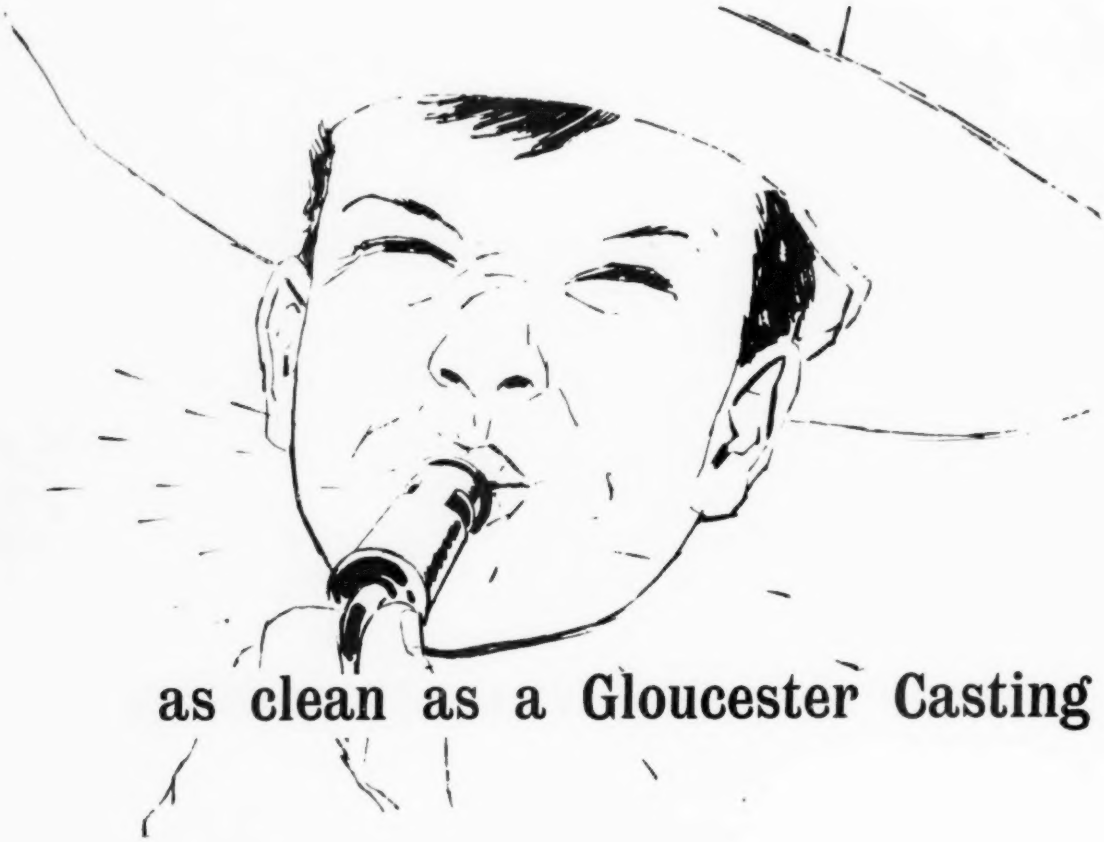

Sunbeam Alpine made by Rootes Group Ltd., Coventry, showing wrap-round, rear window made from 'Perspex' acrylic sheet, by P.D.I. Limited, Birmingham.

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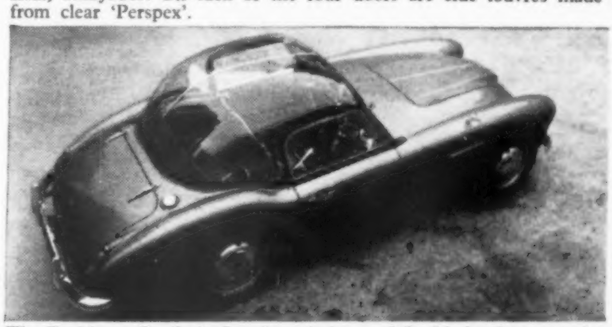

The Pexidome Co. Ltd., London, design and fit this hard top, made from 'Perspex' acrylic sheet, to Austin Healey cars.

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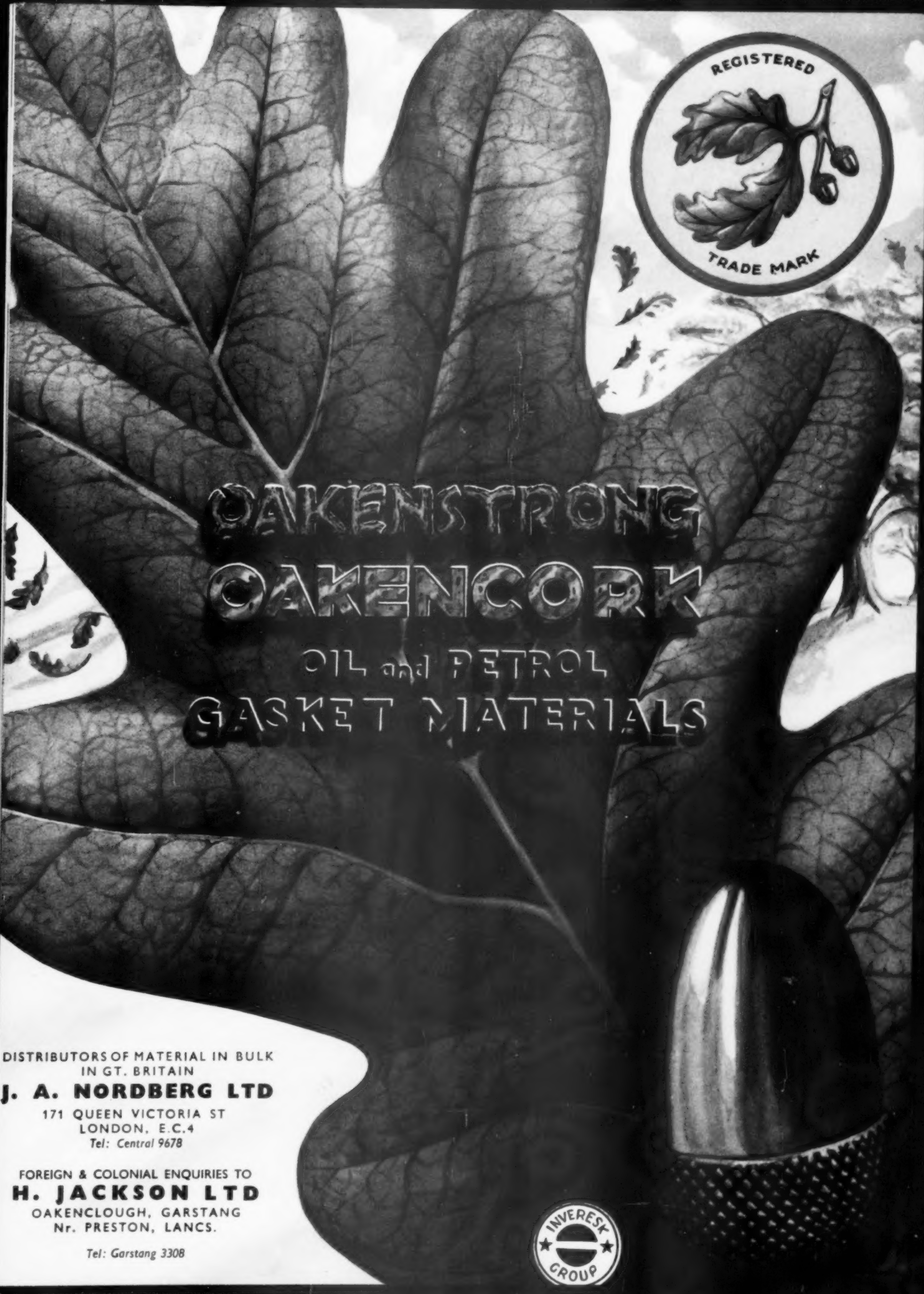
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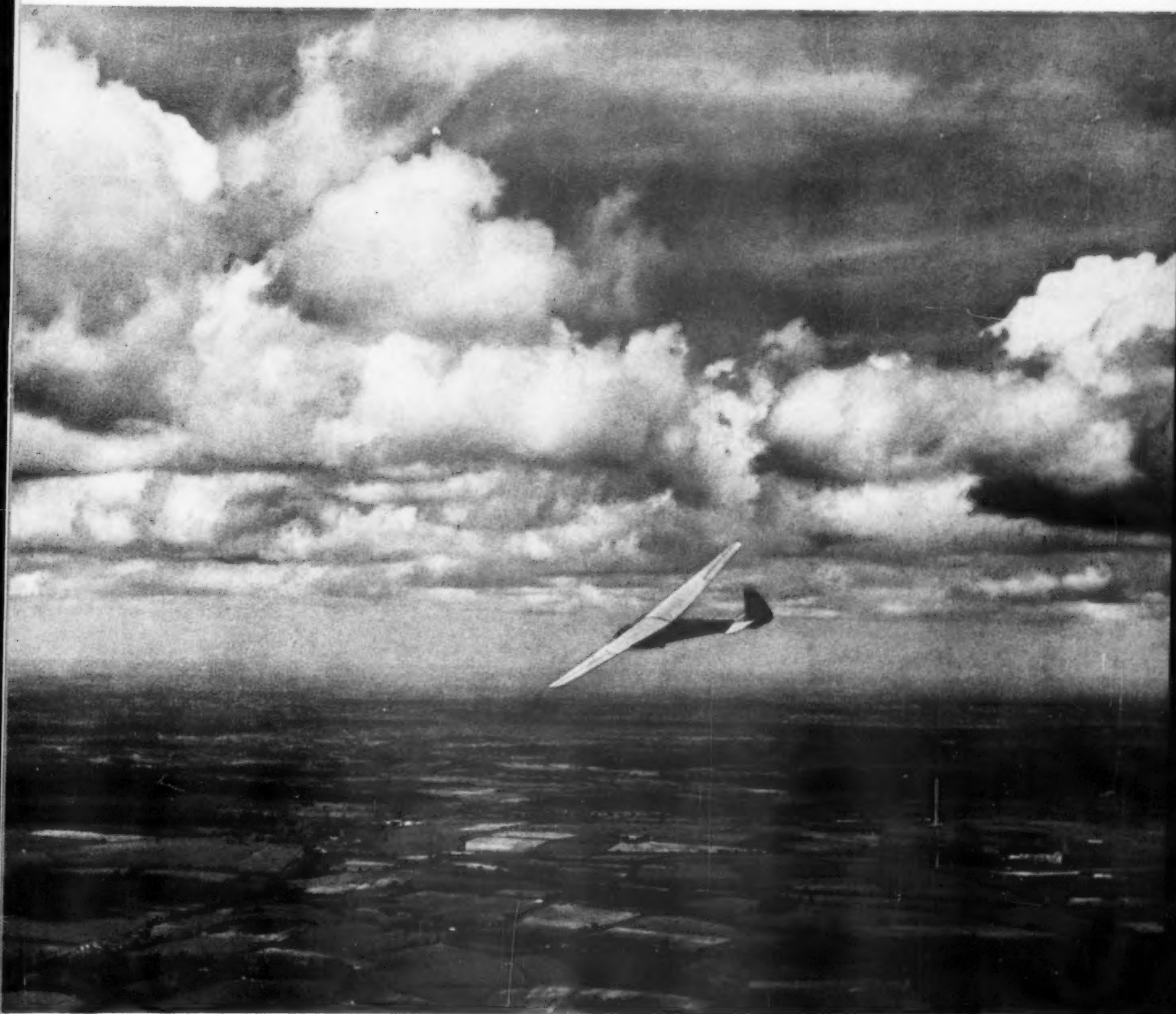

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* *$\frac{1}{2}$ a tenth: .001270 mm. $1\frac{1}{2}$ tenths: .003810 mm. 2 tenths: .005080 mm. 3 tenths: .007620 mm. 5 tenths: .012700 mm.*

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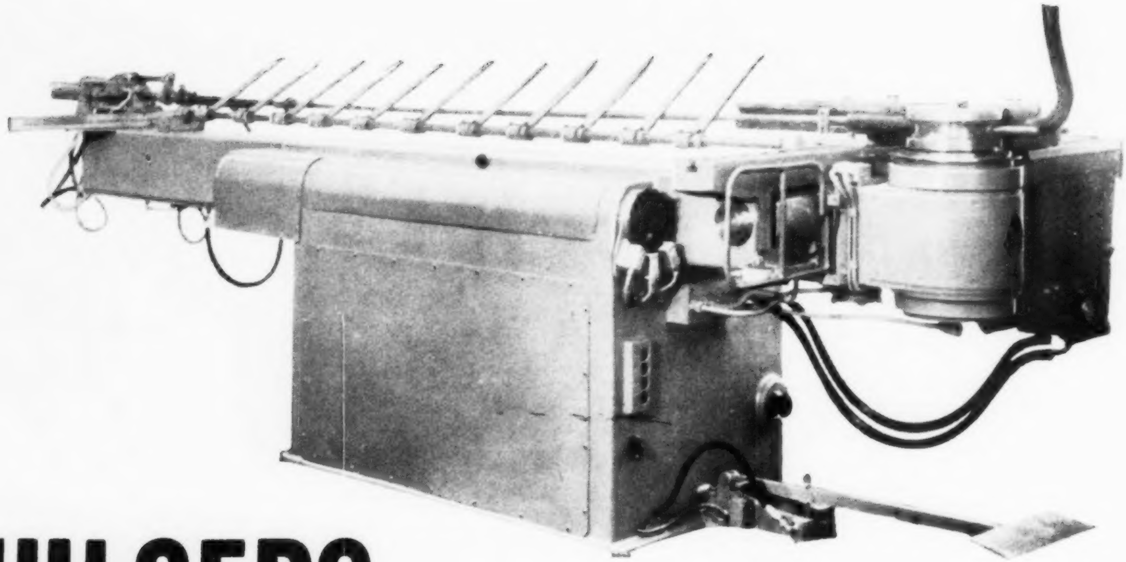

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HDY.114	Semi-Automatic	$4\frac{1}{2}'' \times \frac{13}{64}''$	400
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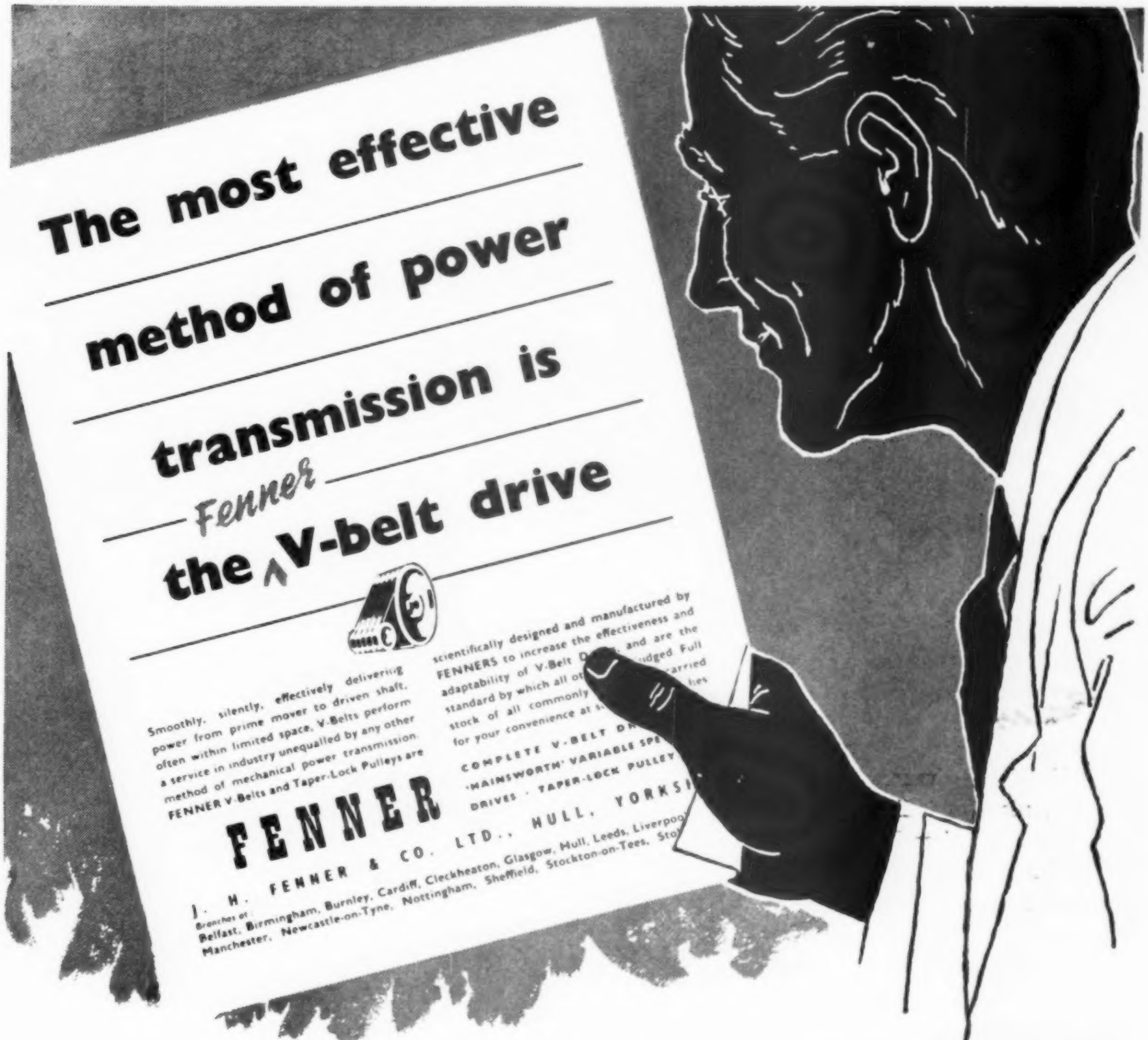

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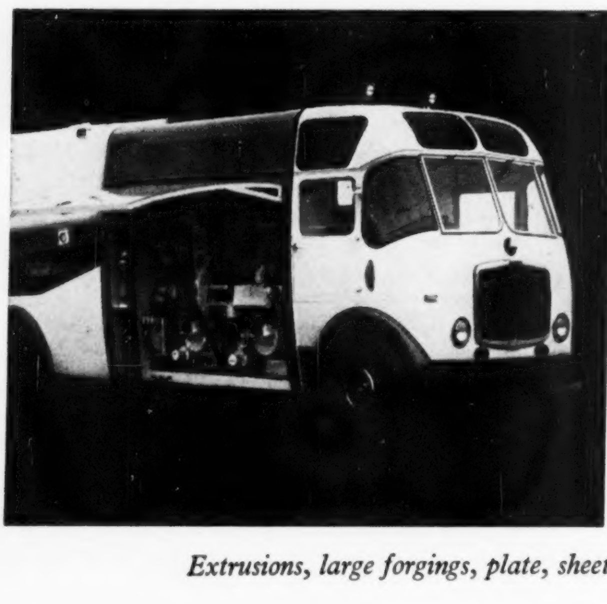

TOP *The 'Yorkshire' Super Fueller, designed and built by Saunders-Roe (Anglesey) Limited, for BP Trading Ltd. is suitable for carrying aviation gasolene or kerosene fuels. Argon shielded arc process was employed in welding the 'MG5' tank structure. It is divided into three compartments, and its overall measurements are 37 ft long, 8 ft 9½ in deep, and 7 ft 10½ in wide.*

LEFT *The Super Fueller is powered by a 165 b.h.p. diesel engine which is also used to pump the aviation fuel at flow rates of over 2 tons per minute. Pumping and metering equipment are housed in the compartment behind the vehicle's cab.*

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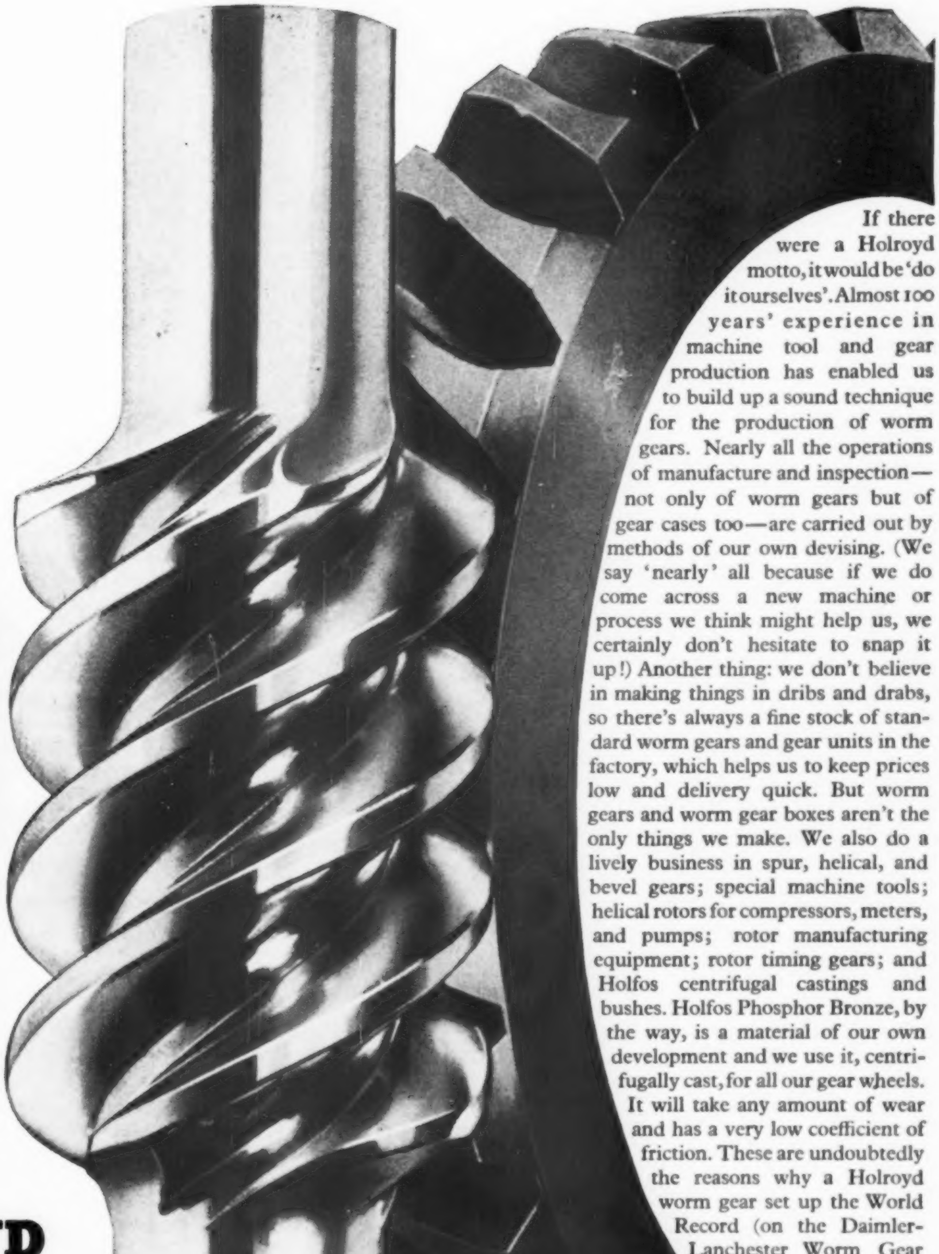

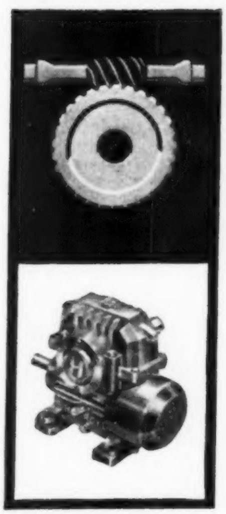

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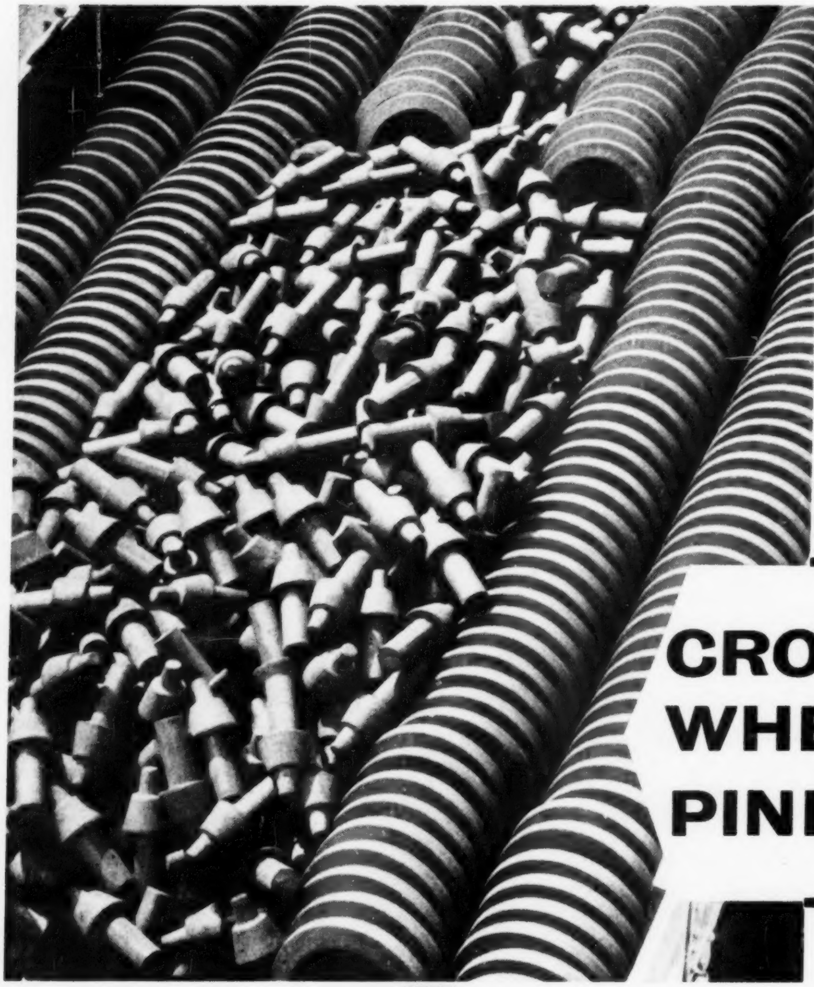

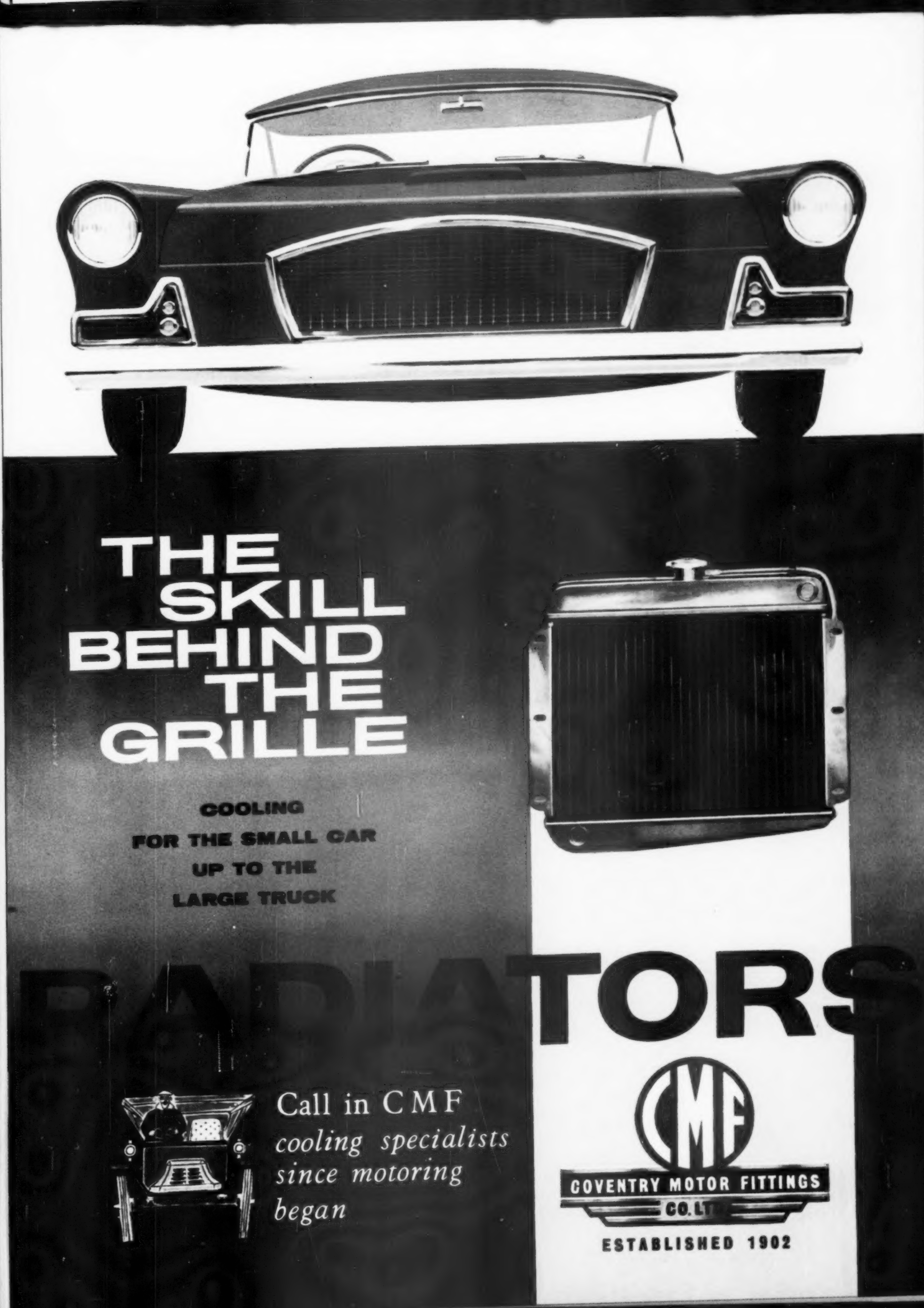
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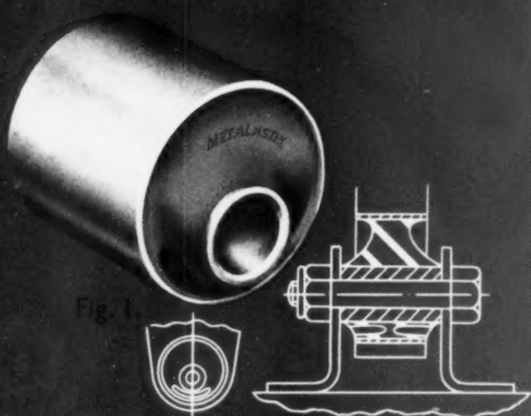


Fig. 1.

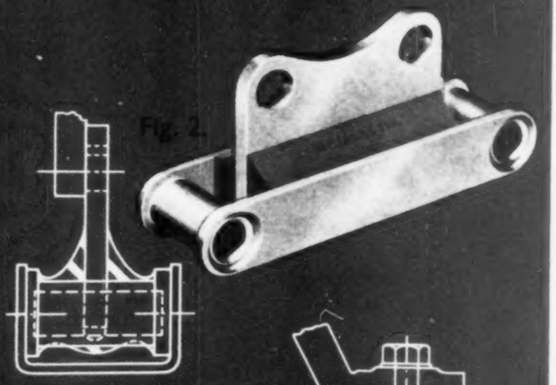


Fig. 2.

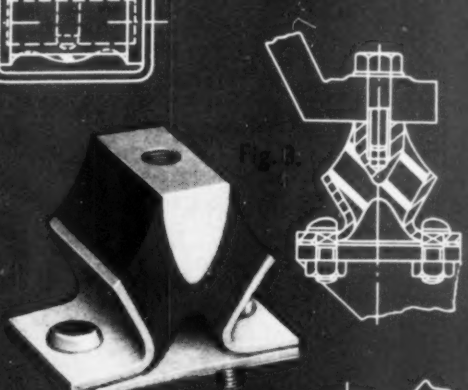


Fig. 3.

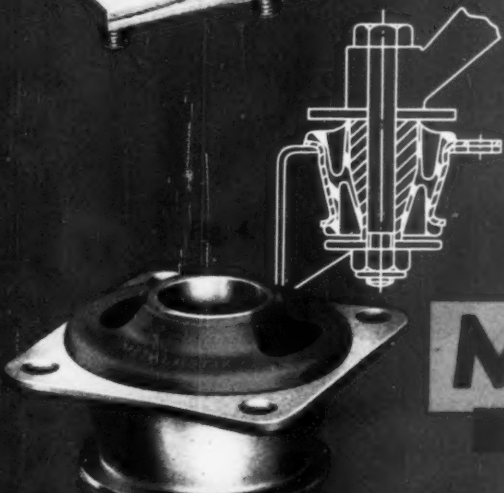
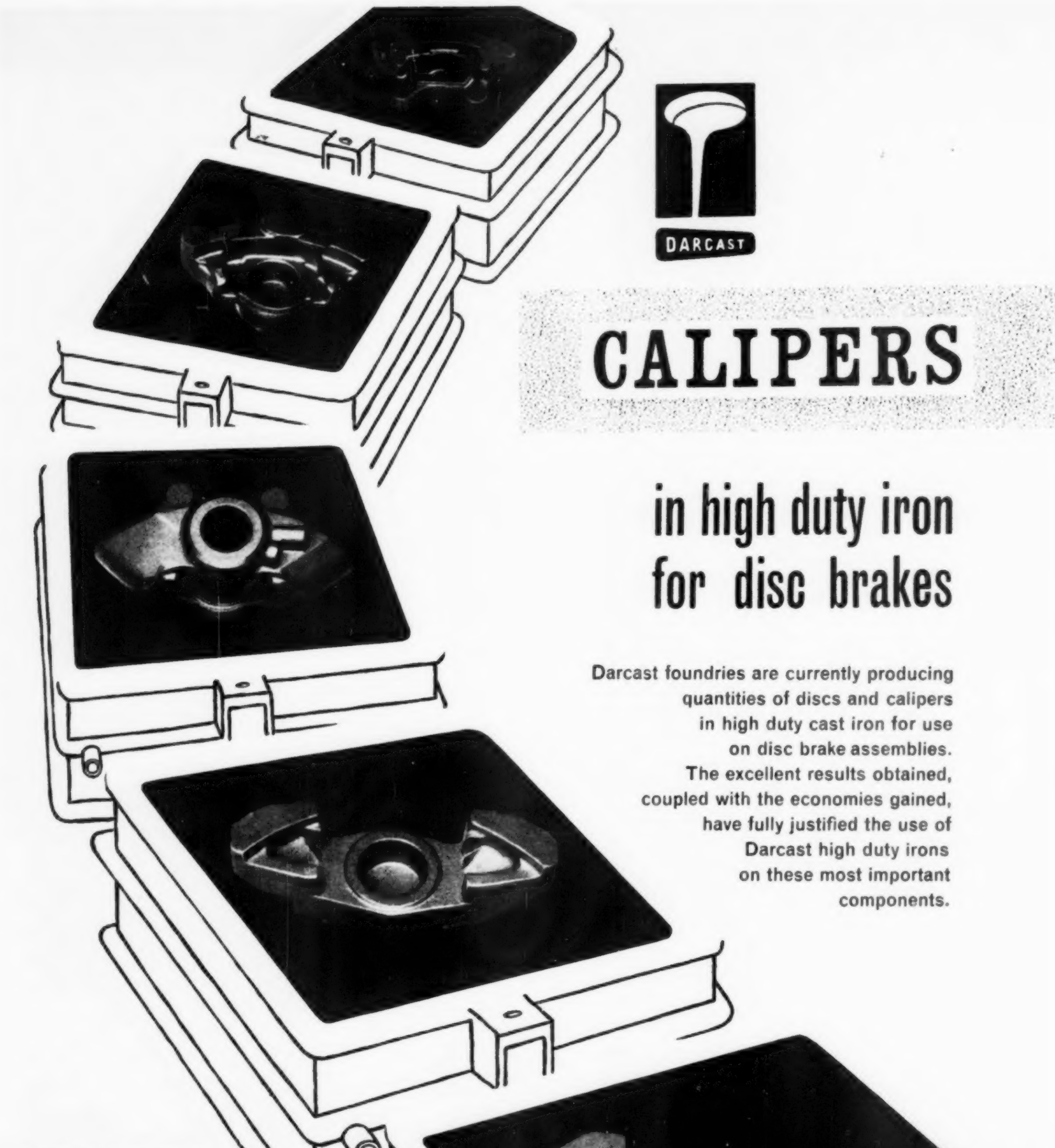


Fig. 4.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AUTOMOBILE ENGINEER

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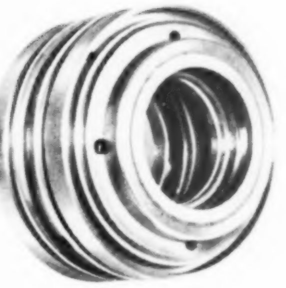

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VOLUME 50 **NUMBER 5** **MAY 1960**

Editor *T. K. Garrett, A.M.I.Mech.E. A.F.R.Ae.S.*
Editorial Staff *F. C. Sheffield (Associate Editor)*
A. Baker, B.Sc., A.M.I.Mech.E.

DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Is Atmospheric Pollution Serious?

RECENT action by the State Legislature of California could have far reaching effects on motor vehicle design. This authority has recently adopted a Bill that, within the next two or three years, will make compulsory the fitting of anti-pollution equipment to exhaust systems. The significance of this news is not only that the legislation will affect cars for export to that State but also that it sets a precedent that may be followed by other governing bodies.

The first question that arises is whether this legislation is necessary; but a governing body responsible for the welfare of 15,000,000 people is hardly likely to adopt such a Bill without having first made a careful assessment of all the relevant scientific evidence—and there is plenty available. Another question that must be answered is whether conditions in that State are peculiar or whether they apply equally in other areas: admittedly, the geographical situation of Los Angeles and its climatic conditions are such that polluted atmosphere tends to stagnate there, with a result that conditions are often very bad, but there are many other cities in the world where fog, polluted to the extent of being injurious to health, is frequently experienced. A survey has shown that, in Los Angeles, one-third of the total contamination by organic material comes from automobile exhaust, which therefore is a significant factor.

The products of complete combustion are, of course, virtually harmless; but where combustion is incomplete, the exhaust also contains carbon monoxide, various hydrocarbons and aldehydes. Of these, carbon monoxide occurs in the largest quantities, and its toxic nature is well known. The complex hydrocarbons and the aldehydes can give rise to respiratory troubles and may be a contributory cause of lung cancer.

Contrary to public opinion, the diesel engine is generally a less serious offender in respect of pollution than is the petrol engine, of which there are also far more in use. It has been found that, if the exhaust of engines with their fuel supply systems in correct adjustment are compared, that of the diesel engine contains lower proportions of both carbon monoxide and hydrocarbons. The harmful effects of tampering with or lack of servicing of injection equipment are familiar, but the petrol engine is at least as sensitive to maladjustment; moreover, the power units of most cars generally are not so well maintained as those of commercial vehicles.

Since satisfactory maintenance cannot be guaranteed by legislation, it is up to the designer to consider what positive steps he can take to ensure reduced pollution. There are two: one is curative and the other is preventive; but so far, both can be made only partially effective. The curative method is to oxidize the polluting constituents after they have left the engine: for this purpose either a catalytic converter or a simple burner is installed in the exhaust system.

Catalytic converters have been proved to be capable of removing up to 90 per cent of the carbon monoxide, up to 60 per cent of the hydrocarbons, and an appreciable proportion of soot, but in some instances the aldehyde content of the exhaust gases is not markedly affected.

This type of unit has the disadvantage that the catalyst requires periodic renewal or regeneration — a converter tested by the D.S.I.R. showed some loss of efficiency after 11,000 miles of operation in a vehicle. Also, since the catalyst does not become effective until its temperature exceeds a certain figure, the device is ineffective under cold-start conditions.

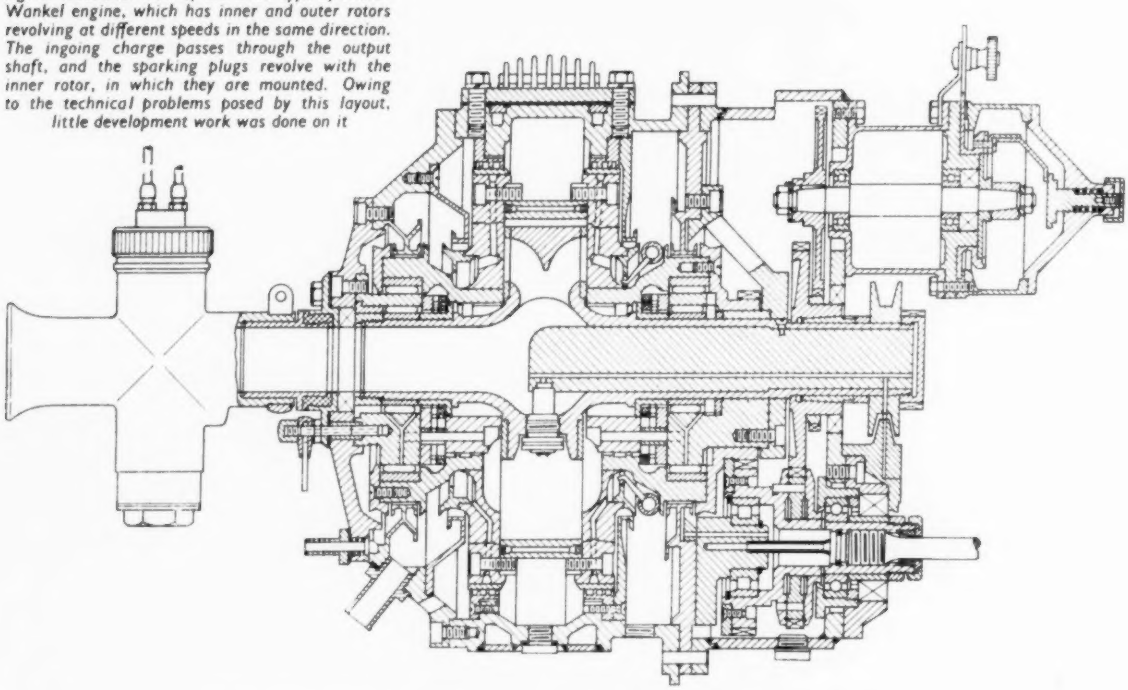
One type of burner, comprising a post-combustion chamber and a heat exchanger, that has been studied in America is apparently reasonably efficient. It is more compact than most catalytic units, and its performance does not deteriorate in use; however, the temperature of the exhaust system is higher.

To prevent the emission of harmful chemicals from the engine itself, and incidentally to improve thermal efficiency, combustion must be made to approach even more closely than hitherto the ideal of the complete burning of all constituents. There are several lines of development that could give worthwhile results. One is the improvement of induction manifolding to give more uniform distribution: this would permit higher overall air:fuel ratios to be used. Attention, too, might also be directed towards the residual exhaust gas, which is a significant factor contributing to incomplete combustion.

Combustion chamber design might be further examined from the point of view of completeness of the burning of the charge, and even more attention than hitherto should be devoted to studying the effects of turbulence. Although there may be scope for improvement without any major departure from current practice, the possibility of evolving alternative systems of combustion designed expressly to give more complete burning should not be overlooked.

Under over-run conditions of operation, the air:fuel ratio of a petrol engine becomes too high for normal combustion, so considerable quantities of unburnt and partly burnt constituents of the fuel are ejected into the exhaust. It follows that the employment of a device, possibly one that is vacuum controlled, to cut off the fuel supply when the throttle is closed with the clutch in engagement and the vehicle in motion would certainly be beneficial in this respect, and it would also reduce fuel consumption.

Fig. 1. Sectional view of the DKM type of N.S.U-Wankel engine, which has inner and outer rotors revolving at different speeds in the same direction. The ingoing charge passes through the output shaft, and the sparking plugs revolve with the inner rotor, in which they are mounted. Owing to the technical problems posed by this layout, little development work was done on it



N.S.U-Wankel Engine

Some Original Thoughts on a Rotary-Expansion Power Unit Worthy of Serious Consideration

R. F. ANSDALE, A.M.I.Mech.E.

A FASCINATING and comprehensive picture of the new N.S.U-Wankel engine has gradually emerged since the first news of it leaked out in November last. It is already abundantly clear that we are not confronted by yet another pipe dream or even wishful thinking, but are indeed witnessing one of the historic events in the long development of the internal-combustion engine. The supremacy of the reciprocating engine has already been challenged in some fields by the combustion turbine and now the N.S.U-Wankel engine presents a further challenge.

It is generally acknowledged that the basis and principles of the gas turbine were known long before modern materials and engineering skill made it a practicable type of unit. Similarly, the N.S.U-Wankel engine is not based upon startling new thermodynamic principles or theory: the novelty of this unit lies exclusively in the brilliance of its general conception and detail design of the components. In this engine there are none of the familiar reciprocating components—they have been replaced by rotating members—and the four thermodynamic cycles, induction, compression, expansion and exhaust, all take place during only one complete revolution of the rotor. The four cycles do not require a full turn of exactly 360 deg, because there is a certain amount of overlap between the exhaust and induction phases, similar to those that occur on two-stroke engines.

As in more advanced two-stroke engines, the gas flow in the N.S.U-Wankel unit may be described as uniflow. This system is conducive to high volumetric efficiency since the gases leaving and entering the cylinder do not have to perform tortuous turns and loops, and there are no nooks or crevices in which inert gas may remain. Moreover, centrifugal forces assist exhaust and may even help induction if axial inlet ports are used. Since there are no valves, only inlet and exhaust ports, the engine must be considered susceptible to induction pressure waves and can, therefore, be tuned in the manner of some two-stroke units.

The new engine represents an achievement that is entirely attributable to a rational and systematic approach to engineering research, and is all the more significant because it was achieved by an eminently practical man. It was only after having completed a systematic statistical investigation that Felix Wankel evolved his new rotary engine. In fact, his success where so many have failed in the long search for an internal combustion engine without reciprocating piston and valve gear assemblies—the limiting factors with regard to power output and speed of conventional engines—can be said to be due to an exceptional ability to separate the wheat from the chaff.

To be precise, N.S.U. have produced two different types of engine: in the first, Fig. 1, the rotor and cylinder rotate in the same direction at different speeds, but this resulted in not only a costly but also a complicated design presenting its own peculiar problems. For instance, the incoming charge passed through the output shaft, moreover, a sparking plug had to be fitted to each rotor flank, and this necessitated slip-ring connections and made cleaning or servicing of the plugs a major operation. It is, of course, possible that the induction through the output shaft might have solved the cooling problem of the rotor, but on balance this engine must be considered a complex piece of machinery. It is called a *Drehkolbenmaschine* or DKM.

Much simpler and seemingly more reliable is the kinematic inversion of this engine, the credit for which must go to Dr. W. Froede of N.S.U. In this, only the rotor and eccentric output shaft rotate at the appropriate different speeds; the induction and exhaust passages are in the stationary cylinder or its end covers. Similarly, the single sparking plug is fitted to the cylinder, where it is accessible, and no slip-rings are required for the ignition. This engine is called the *Kreiskolbenmaschine*, or KKM, and is shown in Fig. 2.

Recalling the rotary compressors used in some German aeroplanes during the last war, it would appear most difficult to translate and describe adequately in simple terms the difference between the DKM and KKM engines. Perhaps the most logical definition is that both engines are rotary-expansion engines, but the DKM has two rotors and the KKM only a single rotor. Since the thermodynamic phases—induction, compression, expansion and exhaust—occur in successive angular positions, there is perhaps a remote analogy with the gas turbine, where the thermodynamic phases are in different axial or radial positions: in the reciprocating piston engine, of course, the phases occur in one place and are separated by time only.

Basic geometric considerations

The unfamiliar epitrochoidal cylinder seems to have captured the imagination of journalists and technical writers alike, and this is perhaps responsible for a certain amount of scepticism among some engineers. Although accurate descriptions of the bore shape have appeared elsewhere, details essential for an effective engineering appraisal have generally been glossed over. The term epitrochoid itself is, of course, not widely understood, and few dictionaries offer a satisfactory practical explanation. However, just a brief examination of the geometric configurations of which the epitrochoid is a particular example will assist in arriving at a more accurate appreciation and even enable the relevant mathematical calculations to be undertaken, if necessary.

In imitation of Felix Wankel's systematic approach, it may be as well to begin with the *cycloid*, Fig. 3, which is the locus traced by a point P on the circumference of a circle while it is rolling along a straight line. If this circle rolls around the outside of another—the base circle—the figure traced by the point P is an *epicycloid*, but if it rolls around the inside of the base circle, in the manner of a planetary pinion, it is a *hypocycloid*, Fig. 4. However, if point P lies neither on the circumference nor at the centre of the rolling circle, the figure traced will be a *prolate* or *curtate cycloid* respectively, Fig. 5, depending on whether the rolling circle rolls round the outside or the inside of the base circle. The latter two curves are also known respectively as *epitrochoid* and *hypotrochoid* curves.

The ratio of base circle diameter : rolling diameter of the N.S.U-Wankel engine must be exactly 2 : 1, and the tracing point P lies on the rolling circle radius. Variations of the locus of P are shown in Fig. 6. Obviously, the waist of the epitrochoid becomes less pronounced as P approaches the centre of the rolling circle because, in the limiting condition, when P coincides with the rolling circle centre, a perfect circle will be traced.

It is interesting to note that Felix Wankel's epitrochoid was obtained empirically. Prof. Othmar Baier of the Technical College, Stuttgart, proved the shape to be an epitrochoid and thus facilitated the mathematical analysis and investigations, which proved of the utmost value in evolving a rational method for machining the bore.

Today Felix Wankel and N.S.U. no longer plot their epitrochoids in the laborious manner described above: instead, they obtain their outline drawings with the aid of an ingeniously conceived draughting machine, as shown in Fig. 7. An annulus gear A meshes with a spur gear B; the diameter ratio $B:A = 2:3$ and the centre-to-centre distance between the two meshing gears is e, while the tracing point P is attached to A by an arm of radius R. If

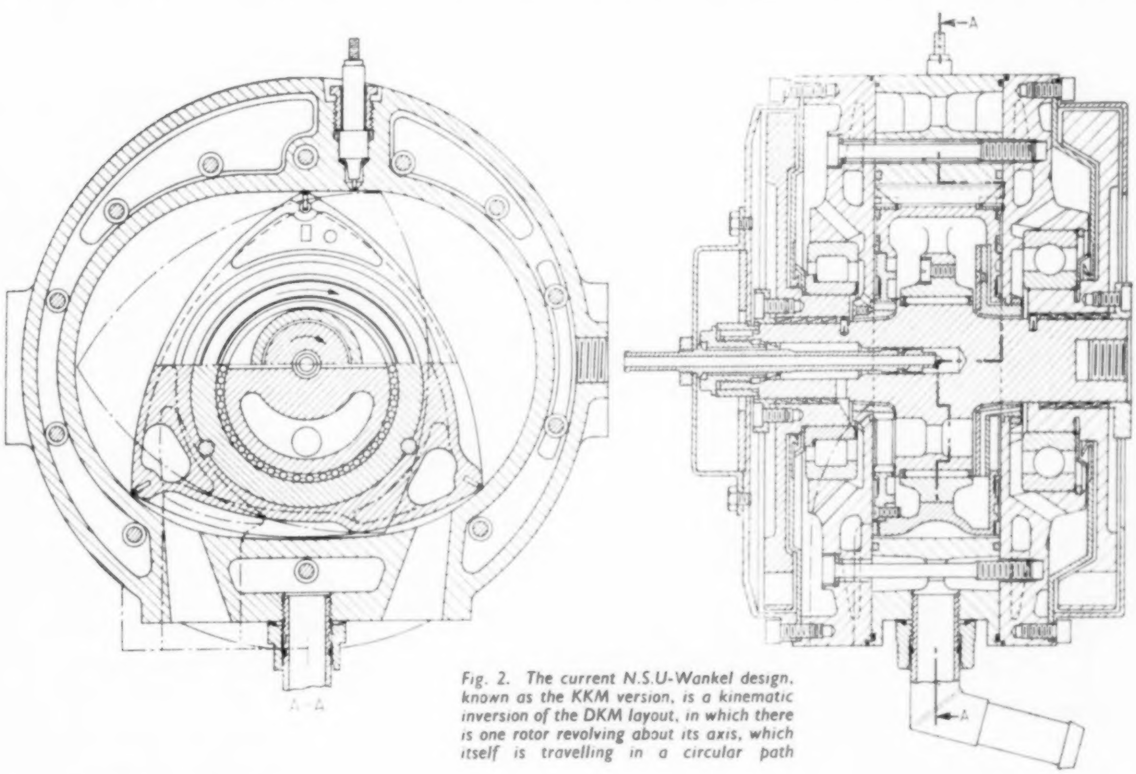


Fig. 2. The current N.S.U-Wankel design, known as the KKM version, is a kinematic inversion of the DKM layout, in which there is one rotor revolving about its axis, which itself is travelling in a circular path

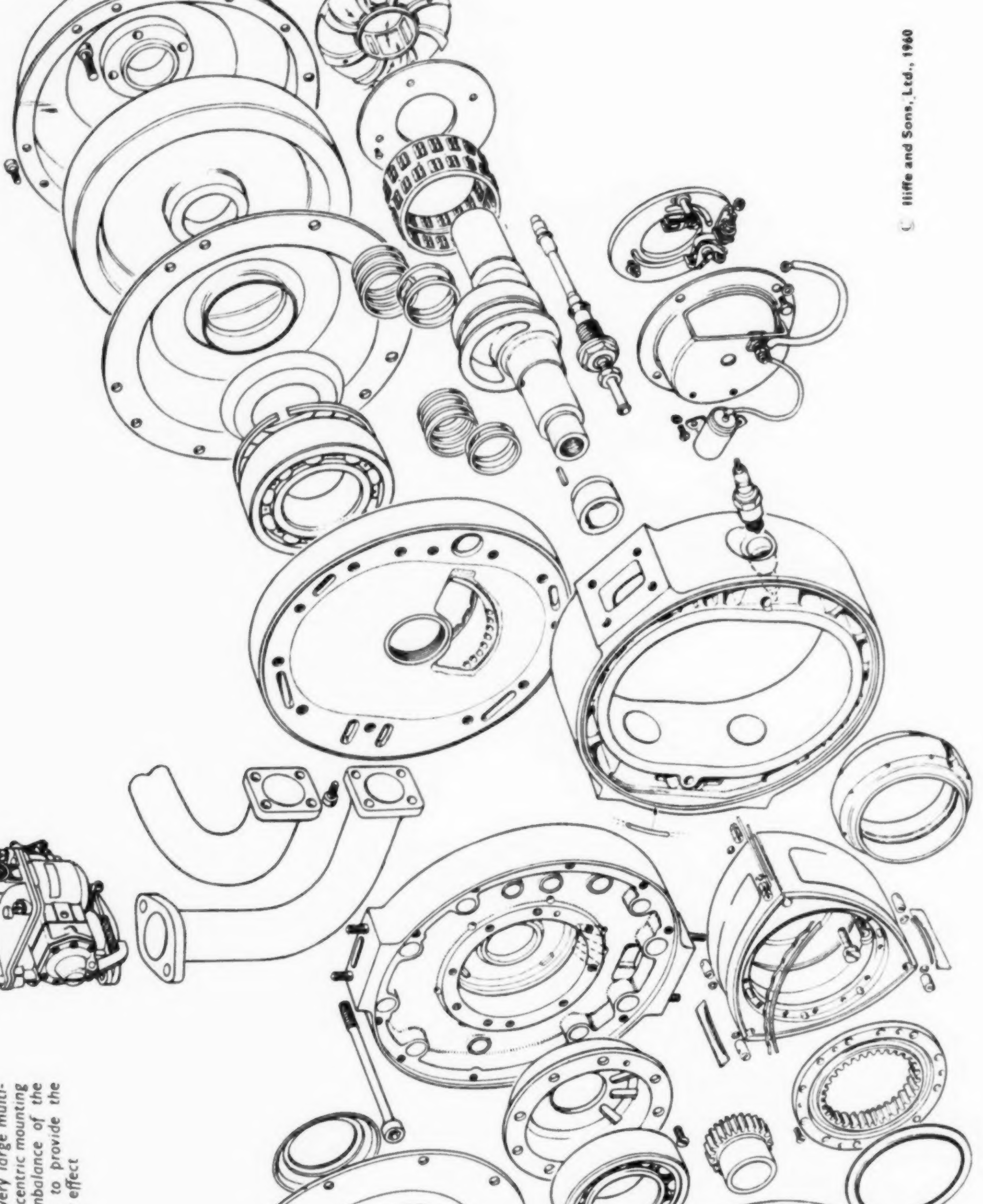
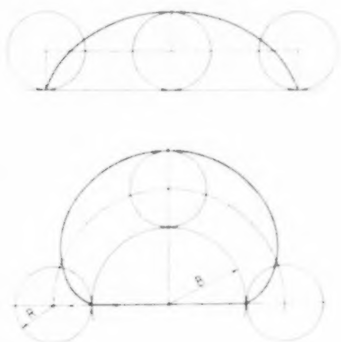


Fig. 16. This exploded view of the later, KKM. unit reveals the constructional details, including the apex- and end-seals of the rotor. The rotor is carried eccentrically on the output shaft by a very large multi-row roller bearing. Because of the eccentric mounting of the rotor and the consequent unbalance of the assembly, the flywheels are drilled to provide the necessary counterweighting effect

the two gears rotate, P will trace an epitrochoid on the platform rigidly connected to gear B. If, however, the tracing point is attached to B and the platform to A, the resultant tracing is a hypotrochoid.

Before going any more deeply into details of the art of generating epitrochoids or of the functional intricacies of the engine, it seems advisable to clarify the terminology and thus to minimize the risk of confusion. With regard to the generated epitrochoid, it is evident that there is a major and a minor axis of symmetry and if the epitrochoid is plotted as in Fig. 5 instead of generated by the machine, these intersect at the centre of the base circle. Since the two axes of symmetry intersect at right angles, they are convenient reference lines.

While the same figure obtained with the generating machine gives the eccentricity e and the radius of generation R, the length of the major axis—bounded by the epitrochoid —is $2(R+e)$ and the length of the minor axis is $2(R-e)$. In other words, the proportions of R and e determine the shape of the epitrochoid: if, for example, the ratio of $R:e=11{\cdot}5$, the epitrochoid will be almost a circle, Fig. 5; and as the ratio $R:e$ diminishes, the characteristic waist of



Figs. 3, above, and 4 show how the cycloid, epicycloid and hypocycloid are produced. Each is the locus of a point on a rolling circle: in the case of a cycloid, the circle rolls on a straight line; for the others, it rolls round the outside and the inside respectively of the figure's base circle

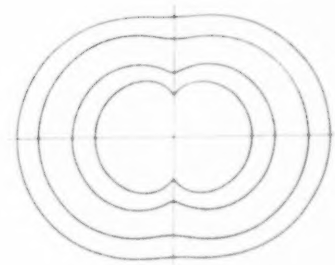
the epitrochoid form becomes more and more pronounced.

The ratio $R:e$ determines the maximum angle of obliquity ϕ between the generating radius R—extended if necessary— and the normal to the epitrochoidal curve. This angle is also the angle at which the apex seals make contact with the epitrochoidal bore. If the outer envelope forms the bore and the inner one the rotor, the contact between the two would always be the same apex lines, which might, therefore, be subject to rapid heating due to friction, and wear could be excessive. The apex seals are the remedy, and Fig. 8 shows the angle of obliquity relative to the true epitrochoid and the actual bore. Although the sealing principles and problems are discussed later, it may be opportune to point out that new apex seals are beautifully radiused, Fig. 8: they quickly wear, or bed down to a yet indeterminate shape and remain thus for considerable periods—up to 700 engine hours have so far been recorded.

The maximum theoretical compression ratio that can be realized with any particular configuration is also determined by the ratio $R:e$, or the angle of obliquity ϕ. Except for considerations in respect of compression ratio, there is no reason why the rotor should conform to the shape of the inner envelope of the epitrochoid; in fact, an equilateral triangle would satisfy equally well the requirements of the rotor.

As already indicated, the angle through which the apex seals swing relative to the normal to the epitrochoid is related to the compression ratio, and Fig. 9 shows some interesting curves, in particular those angles which are applicable to petrol engines and those specially suitable for

Fig. 6. A family of basic epitrochoids, in which the shapes are varied by altering the ratio of generating radius : eccentricity



diesel engines. The assessment is theoretical, as no practical work has been done on the development of the N.S.U-Wankel engine operating on the diesel cycle.

Fig. 10 shows how the curved flanks of the rotor are recessed to avoid throttling the gas flow, at T.D.C., across the waisted part of the casing; this recessing also forms a combustion chamber that promotes burning yet has good anti-knock characteristics. The recess and the minimum volume between rotor and bore together comprise the combustion chamber volume. Numerous recess shapes have already been tried, but the differences in performance are not so pronounced as to enable far reaching basic conclusions to be drawn.

Apex seals

The curved flanks of the rotor do not need to touch the bore either on the minor axis or elsewhere: only the apex seals must always be in contact with the bore. Thus the actual shape of the rotor flanks can be chosen to suit production requirements, provided they enable the prescribed compression ratio to be realized. The apex seals, under the influence of centrifugal force, remain in contact with the bore, since the rotor not only performs a planetary movement round the output shaft centre, but also turns about its own centre. This twofold rotation is responsible for the continuously varying speed of the apex seals: on one of the engines currently being tested the seal velocity ranges between 1,752 ft/min, in the minor axis position, to

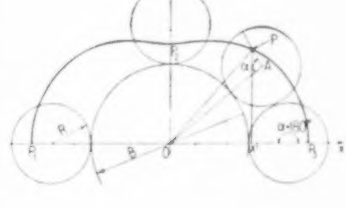
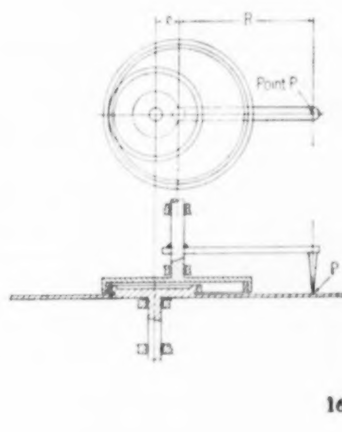
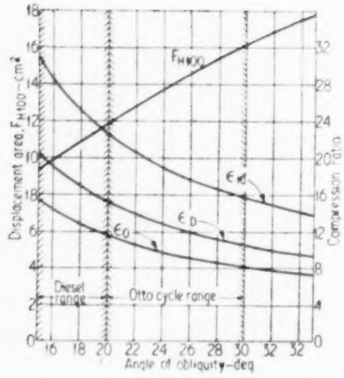


*Fig. 5. If the point **P** lies on the radius of the rolling circle, a trochoid is evolved. In this instance, it is an epitrochoid, as the circle is external to the base circle; a hypotrochoid would be formed if the circle rolled on the inside*

Fig. 7. For the rapid generation of epitrochoids or hypotrochoids, N.S.U. use this ingeniously simple mechanism comprising an annular and a spur gear





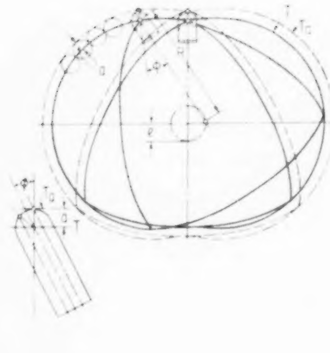
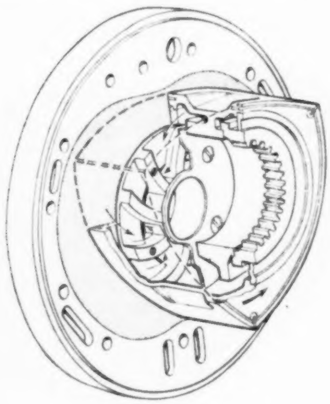


Fig. 8, left: Diagrammatic view of the epitrochoid and the rotor with its apex seal. This seal has a radiused tip to enable it to follow the actual bore, which is parallel to the true epitrochoid. The angle of obliquity, between seal and bore, is ϕ

Fig. 9, extreme left: Graph showing the relationship between angle of obliquity, displacement and compression ratio. ϵ_{id} is the compression ratio curve if the rotor has no cut-out; ϵ_O and ϵ_D are comparable curves for rotors having cut-outs suitable respectively for Otto and diesel cycles

Fig. 11, below: Stationary guide-vanes extract the cooling oil from the rotor. The oil enters through the output shaft

Fig. 10: To minimize throttling of the gas at the waist of the casing, the rotor flanks are recessed, as is shown below. Each rotor illustrated has a different sealing arrangement



4,020 ft/min at the major axis. As a result, these seals are subjected to alternate radial inward and outward accelerations, and the forces due to these accelerations must be added to or subtracted from the centrifugal forces intended to keep them in continuous contact with the bore. Of course, this phenomenon is not peculiar to the N.S.U-Wankel engine, for it is a characteristic of any vane type blower or pump; it is, however, a potential source of trouble.

In the N.S.U-Wankel engine, this problem is not very acute, because the mass of the apex seals is very small, the sealing strips being only 1·6 mm (0·063 in) wide: on vane type pumps and blowers the seals are generally far more substantial. Nevertheless, Dr. Froede showed a cylinder bore which had the characteristic ripple or chatter marks just beyond T.D.C., in the direction of rotation. The traditional remedy for this type of trouble, which resembles the familiar piston ring flutter, is nitriding or chromium plating of the bore, and the provision of suitable expander springs in the slots behind the seals, to assist the centrifugal forces to maintain contact between the apex seals and the epitrochoidal cylinder bore. Indeed, it was the taking of these steps which ensured satisfactory functional engine performance.

From the sealing point of view, another potential source of trouble may be the abrupt change in contact area as the apex seals pass over the inlet and exhaust ports and over the ignition aperture, in particular at the ends where the seals might meet resistance due to their sudden contact with the port edge. Fortunately, that danger has been anticipated and minimized by putting a good radius on these edges. During this development stage and subsequent endurance tests, cylinders made from different materials with variously finished bores were tried—they included light alloy ones with chromium plated bores. Ferrous cylinders have, however, proved the most promising so far: alumina faced apex seals have not been found to be necessary and, therefore, have not been tested.

In comparison with the naturally balanced two-rotor engine—the DKM unit, now discontinued because of its complexity—the much simpler, single-rotor engine is inherently out of balance: although the rotor is perfectly balanced about its own centre, this centre is at the centre of eccentricity of the output shaft. The forces due to this rather large out-of-balance mass can be fully balanced by compensating masses attached to the output shaft, on each side of the rotor, but those due to the cooling oil pulsating through the rotor present another problem unless the cooling passages within the rotor are always absolutely full of oil. Obviously, the oil ought to be under pressure, to eliminate the danger of aeration.

Of all the bearings, it is that of the rotor which operates under the most critical conditions: primarily it must be capable of dealing with the fluctuating combustion impulses and pressures, together with the centrifugal forces caused by the mass of the rotor and the cooling oil within it. The resultant bearing loads are not such as to preclude consideration of plain bearings, which have already been tried; however, multi-row caged roller bearings are currently favoured and will be used until more background knowledge about characteristic engine behaviour and performance has been accumulated.

The planetary movement of the rotor, in an orbit about the output shaft centre, complicates the cooling problem. In fact, it has been found that several b.h.p. were absorbed by the purely parasitic churning of the oil within the rotor. Considerable engineering effort had to be devoted to minimizing these power losses and improving rotor cooling.

Now the coolant enters the rotor, through the hollow output shaft, and it is extracted by stationary guide vanes, Fig. 11 —in shape and arrangement, reminiscent of impeller vanes —which give the oil centripetal velocity towards its return passage. The stationary guide vanes are surrounded by a sheet-metal face sealing ring, not unlike one of those well-known Nylos rings. It is interesting to note that this seal is now fitted rather loosely into the rotor, because previously it had a tendency to distort under the influence of heat and thus failed to function as an effective seal.

Face or axial seals

It has already been said that the N.S.U-Wankel rotary-expansion engine has been the inventor's aim or dream during many years of diligent research and development work on special sealing devices. In 1954, a short time before the engine took shape, Dr. W. Froede published a paper in *ATZ*, in which he reviewed the remarkable progress made with the development of rotary valve engines and, more specifically, he reviewed the state of the art of gas sealing. At that time it was rumoured that N.S.U. had the serious intention of producing rotary valve motor-cycle engines, using the special Wankel sealing elements. Nor was this the first time that Felix Wankel was on the brink of a far-reaching success: according to Dr. Froede's paper, rotary valves and their seals had been developed to such a degree of perfection during the war that the highest authorities were seriously considering whether all German aero-engines should adopt rotary valves in place of the then prevailing poppet valves. This would not only have simplified the engines but also their frontal area would have been considerably reduced. Possibly the impending advent of the gas turbine forestalled this development.

Although it may be intriguing to speculate about these events, it is far more to the point to look at the gas sealing elements developed for these rotary valves, which are illustrated in Dr. Froede's article and which show a remarkable resemblance to those of the N.S.U-Wankel rotary-expansion engine. Sealing arrangements like these may have far wider applications, particularly where the prevailing high temperature precludes the use of rubber and plastics seals.

In Fig. 12, it is shown how the leakage path through a plain split-ring type seal can be practically blocked by a simple insert. Then, Fig. 13 illustrates a complete sealing grid for rectangular ports in a flat or cylindrical surface. Note particularly the link-blocks at the corners: these blocks are intended to seal the leakage paths.

The next illustration, Fig. 14, represents another type of complete seal, consisting of one solid ring and one split ring with inclined mating end faces. Gas pressure forces the rings upward and, because the mating faces are inclined to the end faces, there will be opposing reactions which force the two rings sideways against the recess walls, thus providing a good seal. The small peg prevents relative rotation of the two rings.

A very simple yet sophisticated sealing element for near-rectangular openings is shown in Fig. 15. This consists of two identical piston ring segments, which are sprung into the seal recess, as illustrated. Gas pressure, acting on each free end face, forces the other end to abut snugly against the other ring. With this arrangement, the necessity is avoided of inordinately close manufacturing tolerances. Gas pressure and light corrugated springs, or gas pressure alone, are relied upon to press the complete seal arrangement against the face.

The apex seals of the N.S.U-Wankel engine have already been discussed. However, they present problems that are distinctly different from and perhaps more formidable than those of the axial or face seals, which have evidently been derived directly from the sealing elements just described. These face seals, with their light supporting springs of copper-beryllium, can be seen in the exploded view of the components, Fig. 16. Three arc-shape sealing strips in each face are joined at the corners by link-blocks. These blocks are slotted to accommodate the apex seals, and are thus prevented from rotating, which might cause jamming of the strip seals. However, the primary duty of these link-blocks is to provide completely gas tight seals on each rotor face. For ease of manufacture, the sealing strips and the slots housing them are, of course, circular segments. Nevertheless, they run close to the curved flanks of the rotor.

Rotors with different face seals are shown in Fig. 10. The plate type seal might easily distort under the temperature conditions in the engine, and it is inherently far more expensive to produce. For the strip type seals, various materials have been tested, such as bronze, light alloy, steel

FOUR SEALING ELEMENTS ORIGINALLY DESIGNED BY WANKEL FOR USE IN ROTARY VALVE ENGINES

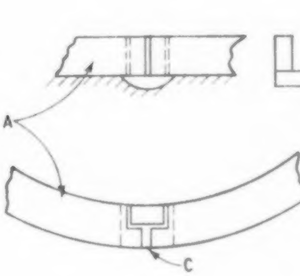


Fig. 12, right: Where a split ring **A** *is employed, a simple insert* **B** *at the gap* **C** *can almost block the leakage path*

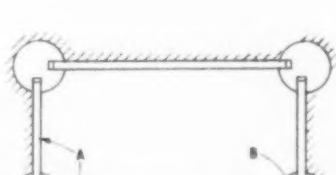


Fig. 13, extreme right: A sealing grid for a rectangular opening. It incorporates sealing strips **A**, *and link-blocks* **B**, *the purpose of which is to seal the corners*

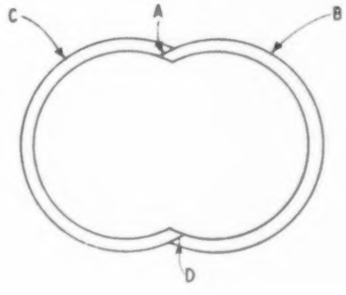


Fig. 15, right: An interesting layout, in which two piston ring segments **B** *and* **C** *are used. Gas pressure, on the exposed end of each, forces them together at* **A** *and* **D**

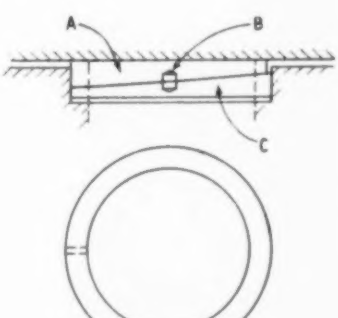


Fig. 14, extreme right: This seal consists of a solid ring **A** *and a split ring* **C**, *with inclined abutting faces. The gas pressure causes the rings to move sideways, thus filling the groove. Relative rotation of the rings is prevented by a dowel pin* **B**

and several grades of cast iron, ranging from the spheroidal graphite type to the high-duty grade normally used for conventional piston rings. So far, the most satisfactory results have been obtained when the latter seals rub against ground-finish molybdenum end faces, this metal having been sprayed on the light alloy end covers. The rotors are now made of spheroidal graphite cast iron, light alloy and fabricated sheet steel rotors having been found wanting in some respect.

From the beginning it has been very difficult to decide whether the N.S.U-Wankel rotary-expansion engine should be considered as a single-cylinder engine, as claimed by N.S.U., or a three-cylinder unit. It may, therefore, be pertinent to attempt to clarify the position. The late Professor Joad would undoubtedly have proclaimed, "It all depends on what is meant by the number of cylinders": indeed it does because, based simply on a count of the number of cylinders or pistons, the N.S.U-Wankel engine is without doubt a single-cylinder engine. Another approach might be to consider a three-node cylinder with a four-sided rotor, which has one extra phase. This phase may be used either to supercharge the engine or make it into a double-expansion unit, in the manner of a compound steam engine; clearly this would remove all basis of comparison unless it were still considered as equivalent to a single-cylinder unit.

Since in the N.S.U-Wankel engine there is one rotor and one induction, compression, expansion and exhaust phase per revolution of output shaft, it may safely be concluded that it is a single-cylinder unit. The capacity of the engine is, of course, the volume contained between adjacent apex seals, the cylinder bore and the rotor flank when a straight line through the points of contact between these seals and the bore is parallel to the minor axis. In this way, it is possible to continue the unfortunate practice of expressing power output relative to the swept volume rather than in respect of the amount of fuel consumed. If the latter basis for comparison were more widely adopted, it would be possible to make more valid and equitable comparisons of orthodox piston engines, all types of rotary-expansion engines and even gas turbines. Moreover, those who prefer to compare power outputs in relation to swept volume could still do so.

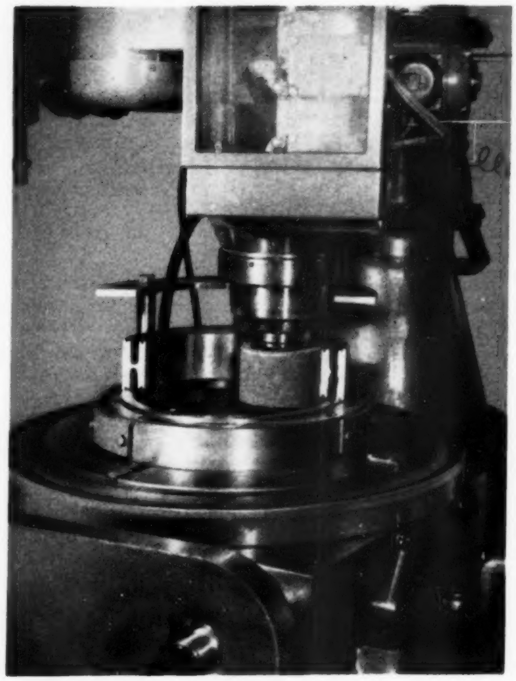


Fig. 18, above: For the production of the actual bore in the casing, a copy-grinding machine is employed. In this view, the machine is set up with a sectioned template and casing. The grinding wheel moves up and down during machining, to avoid any longitudinal irregularities

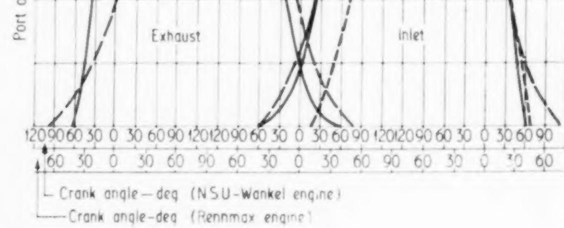


Fig. 21, right: An interesting comparison of the port opening areas of an N.S.U-Wankel engine and of a 250 cm³ racing motor-cycle unit

*Fig. 17, below: Special grinding machine used by N.S.U. to generate templates of epitrochoidal form. On the rotating table **A** is mounted the template **B**; **C** is the eccentric shaft and **D** is the grinding wheel*

In addition to identifying the geometric form of the bore, Professor Othmar Baier suggested the technique of machining this bore with a special form-generating machine. However, it was found expedient to generate only master templates, and to use a copy-grinder for machining the bore itself. In Fig. 17 a template is being generated, and Fig. 18

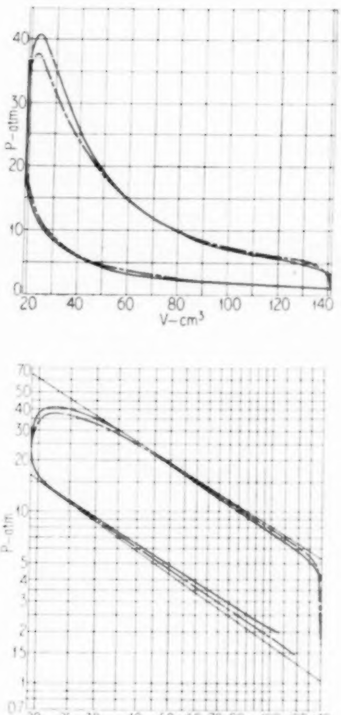


Fig. 22: Comparative indicator diagrams of an N.S.U-Wankel KKM engine, dotted lines, and a piston engine of four-stroke, motor-cycle type; both are of 125 cm³. The lower graph is on logarithmic scales

Fig. 20: A part-sectioned casing, complete with rotor, showing the coolant space, shielded sparking plug, and the position of the ports. The grooves for the O-rings sealing the casing ends can also be seen

Fig. 19: For the accurate checking of the epitrochoid bore, N.S.U. have developed this fixture. Any errors are revealed on the dial indicator

illustrates the copy-grinding machine with a sectioned template and casing in position. The whole grinding head moves slowly up and down while the bore is being ground; this method ensures that there are no longitudinal irregularities of the profile, which might otherwise be caused by imperfections developing in the wheel during grinding.

It is always a problem to decide upon manufacturing tolerances and limits when dealing with non-circular shafts or components of irregular shape. This being so, N.S.U. have found it necessary to develop a simple measuring fixture, Fig. 19. It is a checking device, the construction of which is based upon the principle used in the machine designed to draw epitrochoids. Although no actual figures have so far been released, it is most unlikely that the required profile tolerances are an embarrassment, nor should they raise production costs to unacceptable levels. In practice, it will be found that deviations from the perfectly generated epitrochoid will be of greatest importance close to the minor axis, where they could contribute to a most undesirable flutter of the apex seal; furthermore, it is important that any deviations are not in the form of sudden steps or chatter marks because these could obviously affect performance far more seriously.

In the earlier development models, the major castings were of iron and they had two O-rings in grooves each side of the main central portion: one of these grooves closely hugs the periphery of the bore while the other, a simple circular groove, is close to the outer circumference of the casing, as is shown in Fig. 20. By this means, coolant is prevented from escaping or seeping into the working chamber. It may seem surprising that O-rings can stand up to the prevailing temperature conditions, especially over the expansion and exhaust phase, but some of the later engines, with light alloy castings, have only one O-ring, the inner seal being effected by means of liquid jointing compound. It would appear that only the casing bore and the O-ring grooves present any manufacturing problems.

The two end covers are spigoted to the mid-portion of the casing, and only one fitted bolt or dowel is necessary to ensure correct alignment of the bores housing the eccentric bearings. An interesting feature, already mentioned, is that the light alloy end covers are sprayed with molybdenum on the faces against which the rotor and/or the face seals rub; only the area contained by the epitrochoidal bore is faced and ground.

Since the rotor is a casting, the major part of the machining is on its outer surfaces. Apart from this there are, of course, those operations for machining the sealing strip grooves and the bore that houses the annulus gear and the outer race for the multi-row roller bearing. The eccentric output shaft and the balance weights do not present any unfamiliar production difficulties. This shaft is clearly much simpler than a crankshaft for even a single-cylinder engine, and is considerably stiffer.

Before examining power output, torque and fuel consumption curves, it may be helpful to obtain a basis of comparison by looking at Fig. 21. In this illustration, the port opening areas of a single-rotor 125 cm³ rotary-expansion engine are compared with those of a twin-cylinder

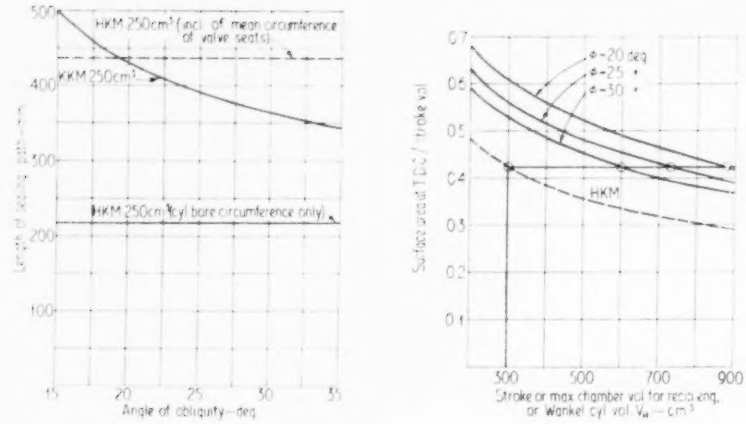


Fig. 23. Two hitherto unpublished graphs of design comparisons between N.S.U-Wankel KKM and piston, or HKM, engines of equal capacity. That on the left shows that the KKM unit is only inferior, with regard to sealing path length, where the angle of obliquity is low. The other indicates the relatively small superiority of the piston engine in surface:volume ratio

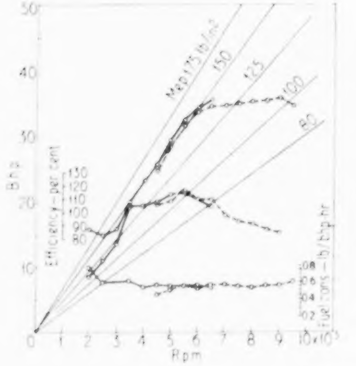


Fig. 24. Curves of the b.h.p., specific fuel consumption and volumetric efficiency of a KKM 250 cm³ engine. The consumption curve of this unit is commendably flat

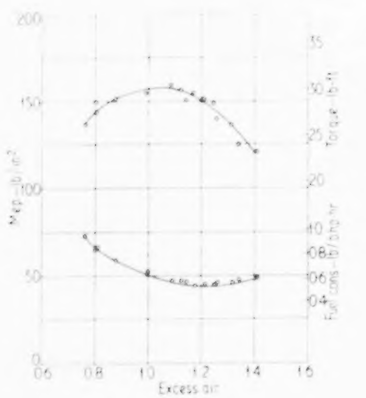


Fig. 25. Interesting curves of torque and fuel consumption of the KKM 250 cm³ unit operated at 5,500 r.p.m., with varied amounts of excess air

250 cm³ racing engine with which the motor-cycle world championship was won in 1954. For specific purposes it would be possible to modify these curves—for instance by shaping the ports in a manner reminiscent of some sleeve-valve aero-engines — but with the N.S.U-Wankel engine little has so far been done in this direction, because this would have upset the basic research programme.

Limitations in respect of available instrumentation, and the complexity of the two-rotor DKM unit, hindered the obtaining of reliable indicator diagrams. However, Dr. Huber of the Institute of Internal Combustion Engines, at Munich, obtained the comparison between an N.S.U. Super Fox four-stroke engine and a single-rotor KKM engine, Fig. 22. The difference between the pressure rise and expansion curves is strictly marginal; although it was expected that the relatively high gas speed over the cooling areas might result in heat losses, these have not materialized. In the *PV* diagram, or more precisely the *log P log V* diagram, it is shown that the characteristic thermodynamic cycles of both engines differ very little and favour, to some extent, the rotary-expansion engine. Whilst this point may be remarkable, its real significance is its indication that N.S.U. has thoroughly mastered the fundamentals of this engine, as is partly illustrated by Fig. 23.

The performance curves of the KKM 250/5 engine, Figs. 24, 25, 26, speak for themselves: the b.m.e.p. of over 150 lb/in² is highly promising for such a revolutionary new design. These curves confirm the presence of induction impulses, or pressure waves. The fuel consumption figures of 0·5-0·6 lb/b.h.p-hr, as read from the commendably flat curve, make an interesting comparison with figures obtained for other engines. For instance, a fuel consumption between 0·52-0·56 lb/b.h.p-hr has been obtained with a 498 cm³ Triumph Speed Twin motor-cycle unit, which develops 27 b.h.p.; and 0·52-0·61 lb/b.h.p-hr with a low-compression Fiat 600 engine, of 633 cm³, developing 22·3 b.h.p. Although comparisons may be odious, or even misleading at times, it cannot be denied that the cumulative know-how behind the design and development of these conventional engines is infinitely greater than that gained during the short few years since the first N.S.U-Wankel rotary-expansion engine was conceived.

First, the basic principles of the design have had to be proved, and so it has not yet been possible to devote much attention to refinements, such as those that ensure long engine life. Nevertheless, several hundred hours of endurance test running have already been successfully completed, and the results of one of the later tests are shown in Fig. 27. The purpose of this full-load test was primarily to determine whether a chromium plated, light alloy casing would have acceptable wear characteristics. A KKM 250/5 engine was used, and it developed a nominal output of 31 b.h.p. at 5,500 r.p.m., which was maintained throughout the test; indeed, it is significant that there was a slight increase in b.m.e.p. towards the end of the test, and that this was accompanied by a slowly falling specific fuel consumption. The wear of the apex seals at the end of this run was measured as varying between 0·0024 in and 0·0044 in, the condition of the seals suggesting that a considerably longer effective life would be obtained. It should also be mentioned that the ignition system, which closely follows motor-cycle practice, has never caused any difficulties. However, it has been necessary to use sparking plugs of a rather higher heat value than would be fitted to an orthodox piston engine of a roughly comparable type.

Curtis-Wright, the American aero-engine manufacturers, have developed larger engines in collaboration and parallel with N.S.U. An engine with a 1-litre chamber displacement is reported to have run several hundred hours without the slightest indication that wear might have a detrimental effect upon performance. Moreover, it may safely be concluded that the sealing and thermodynamic problems of the larger engines are less difficult, so an imposing reduction in the specific fuel consumption may be attainable. Curtis-Wright are said to have advanced so far that a production line is to be put into operation in the near future. Whether aircraft, marine or industrial engines will be produced is not yet known, and, unfortunately, no test data have been released so far.

Any reservations which may have been held regarding the N.S.U-Wankel design are quickly dispelled by a visit to N.S.U. and by Dr. Froede's unhesitant explanations of achievements and difficulties encountered. It cannot be denied that there is optimism at Neckarsulm; however, the case has never been overstated, and both N.S.U. and Dr. Froede insist that it would be premature to release this engine for production in organizations where there is insufficient background and facilities are inadequate for contribution towards further development. For instance, only the two-node epitrochoid, with the three-sided rotor, has so far been explored and, as was mentioned earlier, only Otto cycle engines have been developed. Although these have proved to be surprisingly insensitive to the octane rating of the fuels used, it has not been possible so far to venture into work upon the compression-ignition or diesel cycle. Theoretical aspects of the diesel cycle are demonstrated in Fig. 9.

A brief study of the possible configurations of epitrochoidal rotary-expansion engines and/or compressors shows that there is a wide variety from which to choose; some of these even permit housing of the seals in the casing or outer rotor. However, there is no doubt that considerable research and engineering skill will have to be applied before any of these other configurations can achieve results rivalling those of the N.S.U. and Curtis-Wright designs.

Indications are that, when time permits, Dr. Froede may try the three-node epitrochoid. This will provide one more phase, which can be used either to supercharge the engine or to provide further expansion and thus a compound internal-combustion engine. Multi-cylinder engines—those with more than one rotor—might be developed, and such an arrangement would do away with the flywheels and balance weights used on the present engines. Dr. Froede himself expects that the rotary-expansion engine will first be used for industrial or marine purposes where power and speed variations are limited.

One of the most impressive features of an engine running on the test beds at N.S.U. is the almost complete absence of vibration, which approaches the standard where a threepenny piece can be left standing on edge while the engine is accelerated from tick-over—about 1,000 r.p.m.—to its maximum of almost 10,000 r.p.m and back. The noise level, particularly of the unit as installed in the two mobile test beds consisting of N.S.U. Prinz cars, is well within the limits one expects of orthodox six-cylinder reciprocating piston engines. A short drive in the test cars gave an impression of smooth power and virtually no engine vibration. Acceleration from standstill to about 70 m.p.h. was excellent; gear changes were effected at about 3,000-3,500 r.p.m. and it was noticeable that engine braking was not as effective as with conventional engines. This, however, is as expected because it is impossible to build a low-friction engine which is at the same time an efficient brake. Although exhaust brakes and hydraulic retarders,

Fig. 26. Torque and b.m.e.p. curve of the KKM unit, on full load, plotted against the thermodynamic and output shaft r.p.m.

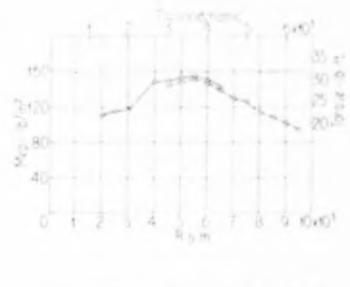
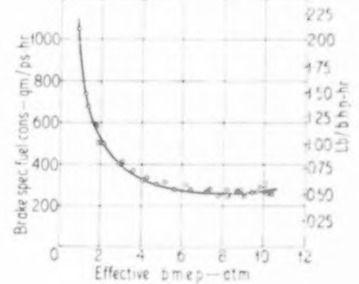
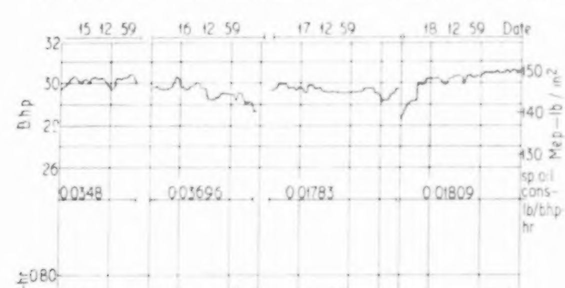


Fig. 28. This curve of part-load fuel consumption shows that, for a b.m.e.p. above about 50 lb/in², the economy is comparable with that of orthodox piston type power units



Behind Dr. W. Froede there are two large scale models: one is of a two-node casing with a triangular rotor and the other has a three-node casing together with a four-sided rotor

Fig. 27. A reproduction of a chart showing figures obtained during an endurance run on the KKM 250 cm³ engine. The test was run at 5,500 r.p.m., at full load. It is noteworthy that both the b.m.e.p. and the specific fuel consumption improved in the later stages of the test



as used on commercial vehicles, come to mind as suitable remedies, it is unlikely that more would be needed in practice than normally good brakes, perhaps of the disc type.

Few ambitious engineers could visit N.S.U. and discuss the rotary-expansion engine with Dr. Froede without wishing for an opportunity to work on this fascinating new design. Obviously, it would be foolish to predict that this engine will completely replace the reciprocating piston engine within a given period. Predictions of this nature seldom mature, since each type of engine finds it own particular field of application. Nevertheless, the part-load consumption curve, Fig. 28, provides some evidence that this engine is far more suitable for automotive applications than the gas turbine in its present state of development.

Improved Lubrication

Notes Regarding Wynn's Friction Proofing, an Interesting Oil, Fuel and Grease Additive of U.S.A. Origin

MOST of the substances currently added to motor oils and greases to improve boundary lubrication act in one of two ways. Either they improve the film strength of the oil, thereby increasing its resistance to local breakdown under load, or they form a low-friction layer of high load-carrying capacity on the rubbing surfaces. In the second category, the best known examples include colloidal graphite and molybdenum disulphide, but the coatings that they provide are physically and chemically distinct from the metallic bearing materials.

An interesting alternative approach to the problem has been made by the chemists of the Wynn Oil Company, of Azusa, California, and is embodied in the range of products known under the general term of Friction Proofing. These substances have the basic property of reacting with the rubbing surfaces to form compounds of low shear strength. Since a rise in temperature accelerates the reaction, the formation of these compounds is most rapid at the points of interference of the microscopic peaks on the surfaces: by virtue of the low shear strength imparted, by these chemical changes, to the peaks, their relative motion causes plastic flow instead of welding and tearing. Also, the compound layer is chemically inert, so it protects the surfaces from corrosion. Since it also has considerable durability, it permits a bearing to run dry for an appreciable period without damage, in the event of failure occurring in the lubrication supply.

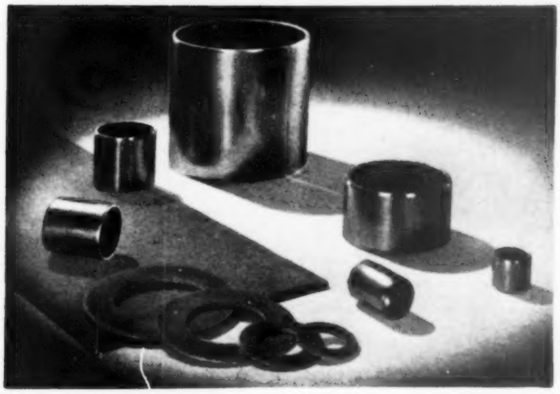
Ready miscibility of the Friction Proofing chemicals with the agent is achieved by the use of a carrier fluid, the nature of which is varied according to whether that agent is a fuel, lubricating oil, grease or cutting oil. Furthermore, other chemicals are added as necessary to furnish a particular grade with secondary properties appropriate to its duty. For example, an engine lubricant operates at higher temperatures than a gearbox or rear axle oil, and is often subjected to oxidation attack; on the other hand, the local pressures encountered in the transmission are extremely high in many instance, so that a greater concentration of the actual Friction Proofing constituents is necessary. In contrast, a product intended for upper cylinder lubrication must be stable at high temperatures and should diminish rather than increase the rate of carbon formation in the combustion chamber; also, it should have a detergent effect on lacquers, to maintain or restore freedom of the piston rings. Each of the numerous Wynn's products, therefore has its own particular purpose and composition.

One of the more recent additions to the range is Diesel Fuel Conditioner. It contains the normal Friction Proofing chemicals, which have their effect on the rubbing surfaces of the fuel pump as well as those in the upper cylinder regions of the engine. Additionally, its specification includes substances to inhibit any polymerization or sludge formation in the fuel during storage, and to minimize the fouling of fuel lines and injectors. An anti-oxidant is also embodied, to give a measure of protection to the fuel storage tank. Improved combustion is stated to result from the greater cleanliness of the injectors and the limitation of the moisture content of the fuel.

The various claims made by the Wynn Oil Company for Friction Proofing appear to be well substantiated by tests by vehicle users, fleet operators and, in the case of cutting oils, on actual machining operations in factories. In this latter use, improvements in both tool life and surface finish of the machined component are reported. The reduction in friction obtainable on automotive applications, investigated over about 200,000 miles on a wide variety of privately owned vehicles, is said to be sufficient to give an improvement of over 10 per cent in fuel consumption. Many of the tests, it is stated, showed a far greater reduction in oil consumption, sometimes as much as 50 per cent, largely because of improvement of the sealing capacity of the piston rings, which resulted from the gradual removal of lacquer.

The various Friction Proofing products were introduced to Europe several years ago, on a relatively small scale, and proved sufficiently successful for the company to decide to go into production on this side of the Atlantic. A laboratory and factory were therefore built in Belgium, where production has been under way for nearly two years. Several of the products for automobile use are available in Britain, namely those for engines, transmissions, upper cylinder lubrication and diesel fuel conditioning. They are handled by Metal Protection Products Ltd., 77A George Street, Croydon, Surrey, who are able to supply further information, including a number of test reports, and to demonstrate the ability of Friction Proofing to fulfil the claims made for it.

GLACIER DRY BEARINGS



The Glacier DU material is manufactured in the three standard forms that are shown here, namely, wrapped bushes, thrust washers and strip

Recent Developments with Materials Based on Polytetrafluoroethylene: Numerous Automobile Applications Made and Many More Envisaged

OVER five years have now elapsed since The Glacier Metal Co. Ltd., of Alperton, Middlesex, introduced their first dry bearings embodying polytetrafluoroethylene, more commonly known as p.t.f.e. The composite material from which these bearings were made was given the designation DP, and its structure and characteristics were reviewed in the December 1954 issue of *Automobile Engineer*. Briefly, it comprised a steel backing, to which was bonded a matrix of porous bronze, impregnated under pressure with p.t.f.e. This DP material was offered in three forms, namely, wrapped bushes, thrust washers and strip.

The main advantages of the use of p.t.f.e. as a bearing material are its low coefficient of both static and sliding friction in the unlubricated state, its resistance to wear and its immunity to almost all chemical solvents. In its pure form, it is attacked only by fluorine gas and by molten sodium and potassium. The disadvantages of poor thermal conductivity, high coefficient of thermal expansion and dimensional instability were overcome in the DP material by the incorporation of the porous bronze matrix, itself a good bearing material.

Although DP bearings proved very satisfactory in a number of applications, their PV factor—the product of the bearing pressure and the rubbing speed—was not very high. As a result, for certain duties, they could not replace existing lubricated bearings without using a larger area. It was soon clear to the Glacier research and development team that a considerable improvement in the PV factor was necessary if p.t.f.e. was to fulfil its potential as a general-purpose bearing material.

A programme was therefore initiated to investigate the effect of mixing various fillers with the p.t.f.e. Laboratory tests of the DP bearings on coated shafts had revealed some improvement in the PV characteristic in cases where the coating was of lead or lead-tin alloy. A test programme on p.t.f.e-lead combinations was therefore put in hand, and the initial results were promising. Consequently, the experimentation with various proportions and methods of adding the lead was continued, and eventually a remarkable result—a trebled PV factor—was obtained. This improvement brought the material into the same strength category, when run dry, as some types of metallic, lubricated bearings.

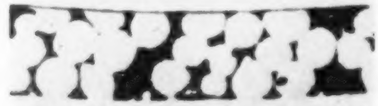
The next stage was to evolve a new range of bearings embodying the improved material. Apart from the replacement of pure p.t.f.e. by the p.t.f.e-lead mixture, these DU bearings, as they are designated, have a similar physical structure to the DP series. The steel backing strip is tin plated to resist corrosion, and the spherical-particle, bronze powder is sintered to it. A composition of 89 per cent copper and 11 per cent tin is employed for this powder, and the matrix has a sponge-like formation, the pores being completely filled by the p.t.f.e-lead mixture. The composition by volume of the bearing lining is 70 per cent bronze, 25 per cent p.t.f.e. and 5 per cent lead. This layer normally has a thickness of 0·010 in. On top of it is an overlay, nominally 0·0008 in thick, of the p.t.f.e-lead mixture, which provides the initial bearing surface.

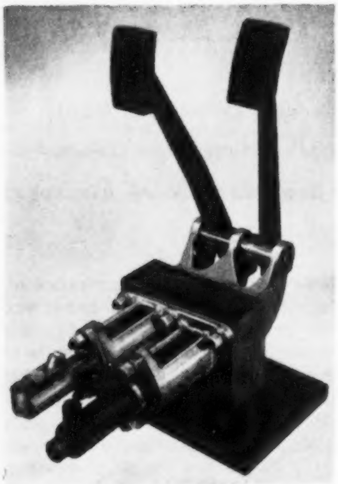
During the bedding-in period of the bearing, which normally occupies about 20 hours, some of the overlay is transferred to the other rubbing surface. This transferred coating is usually about 0·0001 in thick, and it fills any irregularities of the surface and imparts low-friction qualities similar to those of the bearing. A similar thickness of film is retained on the bearing, and the excess is shed as fine flakes, which are quite harmless, provided that they have a means of escape from the bearing itself.

In compression, the yield strength of the DU material is a minimum of 20 tons/in^2. The coefficient of linear expansion of the lining portion only, disregarding the steel backing, is 15×10^{-6} per deg C, and its thermal conductivity is 0·1 cal/sec-cm-deg C. Loading of up to 18 tons/in^2 is practicable, but only in conditions of little or no relative movement. In more normal bearing applications, to avoid fatigue failure, an upper limit of 2 tons/in^2 is recommended.

Because of the wide range of temperature over which p.t.f.e. can be used, DU bearings can operate satisfactorily between -192 and $+280$ deg C. The performance varies little over this range, whereas the DP material showed an appreciable falling-off in load carrying capacity above 150 deg C. Although the bearing itself has good resistance to atmospheric and chemical attack, it may be necessary, when no lubricant is used, to protect the shaft against rusting. Such rusting is not, of course, caused by the DU material.

Photomicrographs of the section through a typical DU bearing before, left, and after running-in; the magnification, as reproduced, is $\times$ 50. The initial surface overlay of the p.t.f.e-lead mixture is virtually removed during the running-in process





Three applications of standard DU bushes. Far left: Control pedals in Aston Martin cars. Left: A similar use in the Jaguar pendant pedals. Below: Another application by Aston Martin, to the gearbox remote control

In general, the life expectation of a DU bearing varies with the PV factor and with the type of duty. However, it has been found that the practicable PV value tends to increase at very high rotational speeds, owing to an air bearing effect. Typical factors for minimum lives of 1,000 and 10,000 hours are given in an accompanying Table. The figures quoted for use against mild steel are applicable also to cast iron, austenitic stainless steel and anodized aluminium, and to mild steel phosphated or electrolytically coated with cadmium, lead or nickel. Although the surface finish is not unduly critical, a good finish has, of course, a beneficial effect on the life for a given value of PV. Copper and high-copper alloys are not compatible with the bearing material.

At very low speeds, between 1 and 20 ft/min, and at loads approaching the maximum of 18 tons/in², the coefficient of friction is at its lowest: in these circumstances it is between 0·05 and 0·10. Within the more normal load range of 10 to 1,500 lb/in², and at speeds between 40 and 1,000 ft/min, the coefficient is from 0·10 to 0·16. This range is similar to that obtained with conventional lubricated bearings, assuming, of course, that the film of lubricant is maintained. The low coefficient at minimal speeds accounts for the absence of stick-slip phenomena. Whereas the bearings are capable of running dry, the presence of a lubricant—including water—usually has a beneficial effect on the coefficient of friction, and substantially increases the PV figures quoted.

A typical wear-time relationship for the DU material is shown on an accompanying graph, which reveals that the wear rate settles down to a low figure after the running-in has taken place. The wear mechanism of a run-in bearing differs from that of an orthodox lubricated bearing. During normal running, if the surface film of the p.t.f.e-lead mixture becomes locally attenuated on the matrix, and a momentary metal-to-metal contact occurs, the heat generated thereby causes the mixture to expand and extrude from the matrix. This extrusion is immediately smeared over the bearing surface, thus terminating the metallic contact. A noteworthy feature of this wear mechanism of the bearing is that it does not lead to any measurable wear of the journal, or other sliding surface.

Over the past four years, extensive comparative figures, on the rates of wear of DU and other dry bearing materials, have been obtained with the aid of twelve test rings. A PV of 16,000 lb/in² × ft/min was chosen and the materials were tested against a mild steel shaft having a ground surface finish of 16CLA, to B.S.1134 : 1950. To avoid variations due to non-uniformity of loading and variation of clearance with wear, thrust washers of the test materials were used, rather than journal bearings. The table of results is reproduced, and it can be seen that, although the test of the DU washer was by far the longest, its total wear was less than $\frac{1}{5}$ that of any other material. In comparison with the DU material, the next best, graphite and lead bronze, has a wear rate more than 63 times as great. The rapid wear of nylon in these conditions is worthy of note.

The main advantages of the DU bearing over other plain bearings, in suitable applications, can be summarized as follows:

1. Dry assembly. Apart from the increased cleanliness that results, there is also a valuable reduction of fire hazard in certain circumstances.
2. Lowered coefficient of friction in either the dry or the lubricated condition.
3. Reduced wear. The average wear rate is about half that of normal, effectively lubricated bushes, and the bearings are more tolerant of dirty conditions.
4. Elimination of stick-slip. With the DU bearings, this characteristic is independent of lubrication, whereas with grease lubricated bearings there is, of course, an increase of stick-slip as the lubricant is squeezed out.
5. Reduced cost. Although a bush of the same dimensions as an orthodox component is more expensive, the higher load capacity permits a reduction in length. Also, the need for greasing holes and nipples is eliminated. The possibility of further saving by design simplification is discussed later.

Bearing types and applications

As in the case of the earlier, DP material, a large range of inch and metric standard bushes is manufactured. All these bushes are of the pre-finished, wrapped type, and the bore sizes go from $\frac{3}{8}$ in up to 2 in, and from 10 mm up to 50 mm. There are 17 sizes in the inch range and 13 in the metric range; for each bore size there is a choice of from two to four different lengths. A schedule of finishing dimensions for journals and housings can be obtained from the manufacturers.

The second standard range is thrust washers, which also are made in inch and metric sizes, with outside diameters between $\frac{7}{8}$ in and $3\frac{1}{4}$ in, and 24 mm and 78 mm. In addition, the material is available in strip form, for the manufacture of slides for machine tools, or other similar purposes. The dimensions of a strip are 4 × 18 in, and there is a choice of four thicknesses between 0·044 in and 0·091 in. If desired,

Table—TYPICAL PV FACTORS FOR DU BEARINGS IN A VARIETY OF NORMAL APPLICATIONS

Type of duty	Mild steel		Hardened steel (540 Brinell)	
	1,000 hr	**10,000 hr**	**1,000 hr**	**10,000 hr**
Thrust bearings	25,000	12,000	30,000	15,000
Slides	12,000	6,000	13,000	7,000
Journal bearings:				
a. Unidirectional loads, load fixed relative to bush	16,000	12,000	25,000	19,000
b. Rotating loads, load rotating relative to bush	25,000	19,000	30,000	24,000
c. Fixed load and oscillating shaft	30,000	23,000	33,000	24,000

the strip can be bent, pressed or embossed, provided that the radius of bend is not less than five times the strip thickness, and that the elongation of the bearing surface does not exceed 10 per cent.

Non-standard products in the DU series include larger thrust washers, of up to 4 in outside diameter, half-round thrust washers of up to double that diameter, and thrust pads. Wrapped bushes can be made to non-standard dimensions within the limits of the standard range. A relatively recent introduction is a range of hemispherical components for ball joints of most types. These are available with internal spherical diameters ranging from $\frac{3}{8}$ in to $1\frac{1}{2}$ in; they can be flanged or unflanged, pierced if necessary, and the p.t.f.e-lead coating can be on either side, although use with the coating on the inside is preferred by The Glacier Metal Co.

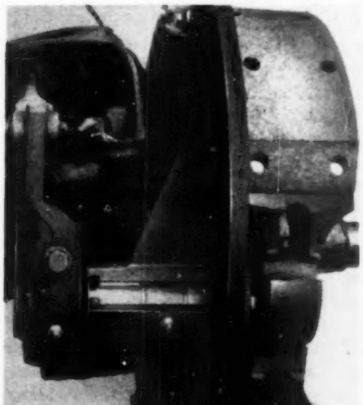
When the DU bearings were first introduced, manufacture had to begin on a relatively small scale, because the market could not be accurately assessed at that stage. Consequently, the unit cost was appreciably higher than that of orthodox bearings. In the earlier stages, therefore, the emphasis was on applications to textile and other machinery, where the price aspect was less critical than it is in the automobile industry, particularly the car section. This policy served its purpose, in that it produced a fund of operational data, helped to offset the development costs and, by the satisfactory performance of the bearings in those applications, prepared the way for the present assault on the automobile market.

In spite of this deliberate canalization of activity, several vehicle manufacturers were attracted by the possibilities of the DU material and have already standardized it in a variety of applications. For example, on the Humber Hawk and Super Snipe cars, the use of DU thrust washers in the king-pin assemblies has effected a marked reduction in the steering effort. A DU bush is employed in the gear change linkage of these cars. Another use of the bushes is as pivot bearings for control pedals in Jaguar and Aston Martin production cars; in the second of these makes, the bushes are also used in the throttle linkage and the gearbox remote control.

Other Aston Martin applications are to the tensioner sprocket of the chain drive to the camshafts and, on the Formula 1 racing car, to the pivot bearings of the front suspension upper wishbones. On the Standard Vanguard fitted with Borg-Warner automatic transmission, the gear change cross shaft is carried in DU bushes. The 1959 Lotus Formula 1 and 2 racing cars embodied an unusual high-speed application—a steady bearing for the middle of the propeller shaft. Although the loading on this bearing was relatively light, the $1\frac{1}{8}$ in diameter shaft revolved at up to 8,000 r.p.m., giving the high maximum linear velocity of 2,360 ft/min.

Hitherto, the commercial vehicle companies have probably shown a greater interest in the DU material than have the car manufacturers. There are several probable reasons for this, of which the cost aspect has already been mentioned; by virtue of large-scale production and the ability to reduce bearing length, this objection has now been overcome. Also, operators of commercial vehicles are more cost-

At present, there are more applications of DU bearings on Scania Vabis commercial vehicles and buses than on any other make. DU bushes are used at the brake camshaft pivots, left, the brake and clutch pedal pivots, centre, the brake shoe heel pivots, right, and at other points



On some Scania Vabis chassis, the steering king-pin swivels in DU bushes; thrust is taken on a DU washer housed in the bottom cover of the assembly

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In addition to these applications to vehicles and engines already in production, there are several others which have been recommended for adoption by the vehicle manufacturers concerned, but which have not yet gone into production. Naturally, it is not possible to be specific about these at the present juncture, but they include king-pin assemblies, a gear change mechanism embodying a hemispherical joint, and transmission bearings. As an indication of the rapidly increasing interest in the DU material, over 40 test projects are now being carried out by car, commercial vehicle, motor-cycle and component and accessory manufacturers. These projects cover a wide range, including suspension and steering ball joints, also bearings for suspension arms, carburettors, starter motors, gearbox and other transmission duties, and fuel injection pumps, in addition to further examples of the applications already mentioned.

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Another line of development, and one that the company considers to be entirely feasible, is the elimination of lubrication of the rocker gear of overhead-valve engines. The conventional bearings in the rockers could be replaced by DU bushes—they have already been tested in that duty—and the valve guides and push rods or rocker ends provided with a p.t.f.e-lead bearing surface, as will be described later. Lower cost should result from the deletion of oilways in the various components. Another advantage would be a reduction in oil consumption, particularly in the case of a worn engine: the amount of oil lost down the inlet valve guides is often considerable, and this loss is not only uneconomic but it also gives rise to heavy deposit formation on the valve heads and in the combustion chambers.

Other dry bearing products

Clearly, there are numerous bearing applications in which a pre-finished component is not acceptable, because the effective bearing surface must be machined. Also, there are shapes and sizes, particularly small ones, that cannot conveniently be formed from a composite strip. To cater for such instances, Glacier DQ material has been evolved; it has most of the performance merits of the DU type with the additional advantage that it can be readily machined or moulded, but it is more expensive.

This DQ material is made from p.t.f.e., graphite and spherical bronze. The bronze is in powdered form and is mixed with the other constituents. Since there is no steel backing, and the bronze does not form a matrix, the mechanical strength of DQ is lower than that of DU. The ultimate tensile strength and the shear strength, both measured at 20 deg C, are respectively 875 and 2,000 lb/in^2. Considerable variation occurs in the compressive strength with temperature: at 20 deg C, the strength is 2,700 lb/in^2, but it rises to 8,300 lb/in^2 at -196 deg C and falls to 1,900 lb/in^2 at 100 deg C.

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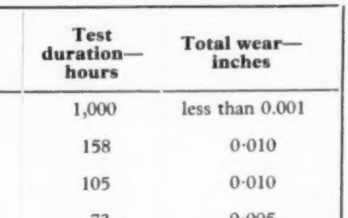
Right: Graph of wear against time for DU material. Below: In this table are given the results of wear tests on a number of types of dry bearing



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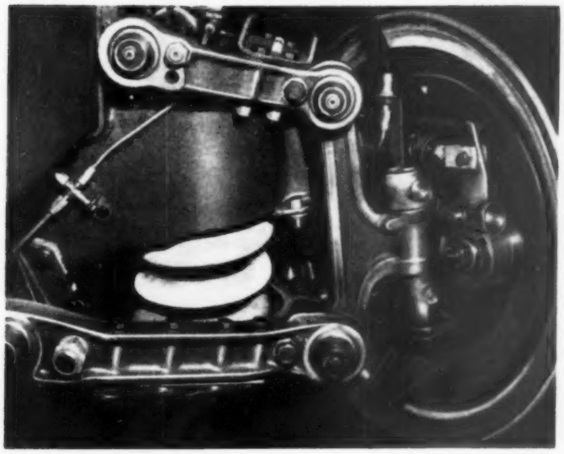
Phenolic resin with MoS_2	73	0·005
Graphite, bearing grade	24	0·005
Woven asbestos impregnated with resin and MoS_2	0·8	0·005
Nylon	0·3	0·010



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While the normal DQ material has good resistance to chemical attack, a modified version is available that provides almost complete inertness. This alternative material is known as DQ2, and its greater chemical resistance is obtained by the use of a non-metallic filler, with some consequent loss of mechanical strength and wear resistance. Because of this, the makers do not regard it as suitable for automobile applications.

The DQ material is available from stock in bar and tube form. There are ten sizes of bar, ranging from $\frac{1}{4}$ in to 2 in diameter, and six sizes of tube. The smallest tube has an outside diameter of $\frac{3}{8}$ in and a bore of $\frac{1}{4}$ in, and for the largest the corresponding dimensions are 2 in and $1\frac{5}{8}$ in. In addition, special components, including mouldings, can be produced by Glacier to individual requirements.

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For the surface treatment of parts on which a low-friction, wear-resistant surface is required without lubrication, the company has evolved two processes, known as DG and DL. Both consist of the application of a thin, adherent coating to a prepared surface, but only the DG, in which the coating has a p.t.f.e. base, is thought to be suitable for use on motor vehicles. Its coefficient of friction is lower than that of DL, and its wearing qualities and chemical and corrosion resistance are superior. Also, it has better mechanical properties and some tolerance for liquids, and therefore can be used to enhance the effectiveness of normal high-pressure lubricants. The main advantages of DL are its resistance to atomic radiation and its ability to withstand higher operating temperatures. It is a molybdenized material and tends to be washed off by liquids.

The first stage in the DG process is the chemical treatment of the parts to give them a good keying surface. A coating of the p.t.f.e-base mixture is then sprayed on at room temperature, and the coated parts are subsequently heat treated to develop the necessary adhesion. It is, of course, essential to ensure at the design stage that this heat treatment will not affect the strength of the parts.

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Because of the thinness of the coating, the mechanical and physical properties of the treated item can be regarded as those of the parent metal. However, the coating can be scratched relatively easily, so care is needed during storage and handling; also, the material is not suitable for operation in dirty or abrasive conditions.

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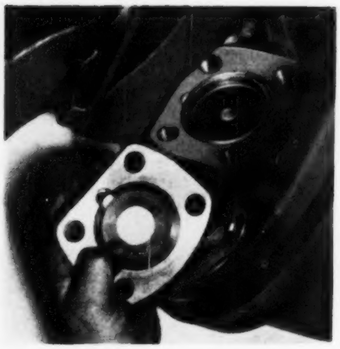

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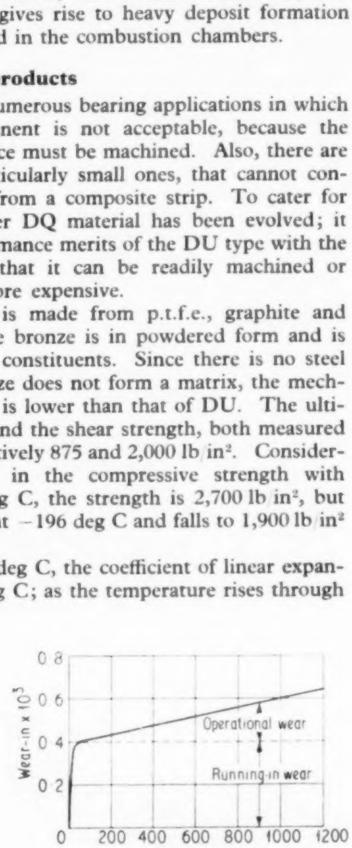

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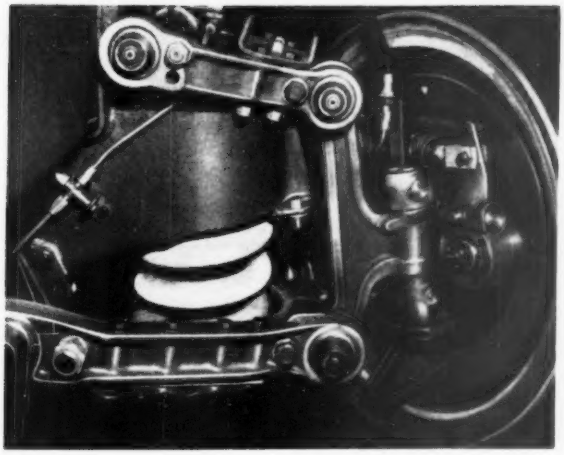

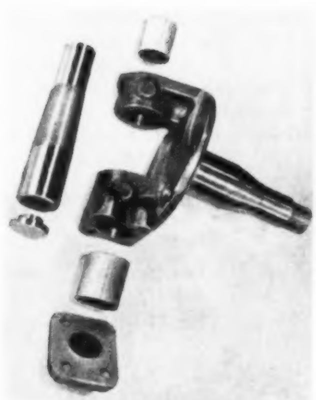

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Components of the Hydrosteer fully integral, power-assisted steering gear. Just above the camshaft can be seen the parts of the valve, valve chamber and sealing assemblies. The casing and its top and end covers are LM22 aluminium alloy castings

HYDROSTEER POWERED STEERING

Integral System for Larger Cars; Cam-and-Peg Mechanism and Spool Type Valve; Marked Ratio Variation from Straight Ahead to Full Lock

UNLIKE previous power-assisted steering systems produced by Hydrosteer Ltd., of Luton, this company's latest unit is of the fully integral type : as in the case of the Burman and Saginaw designs, described in the February and October 1959 issues of *Automobile Engineer*, there are no external jacks. The cam-and-peg mechanism employed is of largely orthodox layout, except in that the cam is surrounded by the skirt of a piston working within the body of the steering box; the hydraulic power is transmitted directly from the piston to the peg carrier. A hydraulic valve, of spool type, moves axially in response to steering wheel movement to cause oil pressure to be applied to the appropriate side of the piston. The equipment has not yet been adopted for fitting to production cars, but it is being tested by several manufacturers.

From the outset, it was decided that the fail-safe type of design should be adopted : in other words, the driver should still be able to retain full control of the car in the event of a failure of the power system. Another essential was that jamming of the valve, either under very heavy loading or because of foreign matter in the oil, should be made virtually impossible. A third primary requirement was a high speed of response of the hydraulic assistance. In the design adopted, the valve can always keep pace with the speed of operation of the steering control, though an inadequate pump output can result in the full degree of servo not being available for exceptionally rapid wheel movements.

Rapid response is, of course, only one desirable feature of a powered steering gear : the response must also be sensitive and progressive if lack of "feel", a point of criticism of certain earlier designs, is to be avoided. The Hydrosteer technicians have aimed at giving the unit a feel comparable with that of a good unassisted system. They claim to have obtained this by careful attention to the degree of feedback, and to the design of the seals and the springs of the hydraulic valve. These springs are of disc type, with a high rate, and are balanced in their neutral position. Other important considerations in the design were that the size and weight should be the minimum consistent with adequate strength, and that the valve assembly should be of a simple type in which wear and its attendant backlash would be restricted.

To obtain the maximum benefit from power-assisted steering, it was felt desirable that the ratio should vary appreciably between the straight-ahead position and full lock on each side. Obviously, a relatively high overall ratio was desirable to avoid excessive steering wheel movement during low-speed manoeuvring. However, in view of the low steering effort attainable, too direct an action in the middle of the range could result in the vehicle's being subjected to dangerously large lateral forces at high speeds. The cam selected therefore provides moderate gearing in the region of the straight-ahead position and a progressive numerical fall in the ratio towards the locks. On a typical example, the ratio diminution from the middle to full lock in each direction is about 39 per cent.

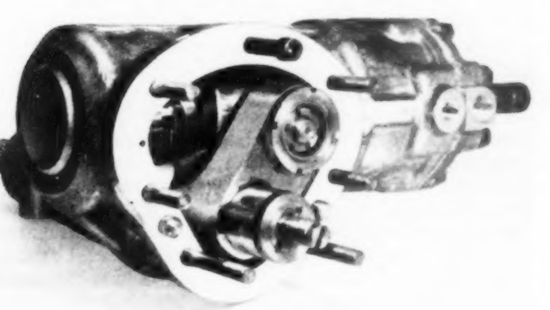
The nominal maximum working pressure is 1,000 lb/in^2, but there is usually sufficient power available at 700 lb/in^2. In the event of failure of the power supply, only 0·6 in movement of the steering wheel rim is needed, in either direction, before positive mechanical transmission occurs. Such a small amount of backlash should be undetectable under normal operating conditions. A typical Hydrosteer

unit has length, breadth and height dimensions, as drawn, of $11\frac{31}{32}$ in, $5\frac{7}{16}$ in, and 14 in, though the last figure varies with the length of rocker shaft specified; an average dry weight is quoted by the manufacturers as about 20 lb.

Design and construction

The main housing and its three covers are substantial castings in LM22 aluminium alloy. A hole in the lower end of the body accommodates the spigot of the bottom cover, which is pushed home, from within, against a shoulder; oil sealing is effected by means of an O-ring. Contained in a counterbore in the cover are a p.t.f.e. oil seal, of castellated section, and the channel-section outer race of the lower of two uncaged ball bearings that support the camshaft. The upper end of the cover is relieved circumferentially to provide a spigot for the sleeve in which the piston operates. This sleeve is of cold-drawn seamless tubing, to B.S.S. 980/CDS2; it is a push fit in the bore of the housing and, since it is clamped by the components above it, prevents any upward movement of the bottom cover. The wall thickness of the sleeve is 0·109 in, and near the upper end is a kidney shape hole, which provides operating clearance for the peg assembly on the end of the rocker arm.

Thirty-two balls, $\frac{5}{32}$ in diameter, are used in each of the shaft bearings just mentioned. They run directly on the camshaft, which has a diameter of $1\frac{3}{8}$ in for most of its



The inner castellated ring-nut on the rocker arm is for adjusting the cup-and-cone bearings of the peg. At the far end of the casing are two tapped holes, which are for the unions of the oil supply and return

length and is machined from En.34 steel; this component is case-hardened on the cam track and on the bearing tracks. The shaft and the piston, which is of En.32B steel, have a ground finish on their working surfaces, whereas the bore of the sleeve has a special fine finish. Although the piston is supported in the sleeve for its full length, the presence of the cut-out makes it necessary for the external piston-ring type seal to be at the upper end. To minimize friction, the bore of the piston does not bear on the shaft, the only contact being by the internal hydraulic seal. This seal is of unusual design: it comprises a D-section p.t.f.e. ring, disposed with the curved surface bearing on the shaft, lightly loaded inwards by a square-section, synthetic rubber ring installed around it in the same groove.

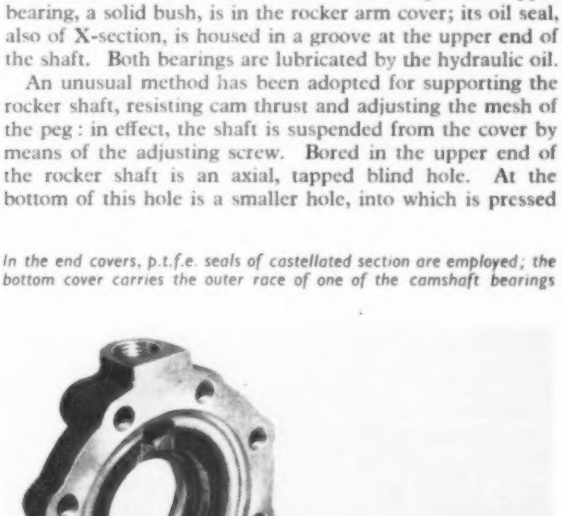
The cam track has a section somewhat similar to that of an Acme thread. Because of the geometrical considerations, the track is not a regular helix but varies in pitch, included angle and depth. The radius of swing of the follower peg is greater than the distance separating the axes of the rocker shaft and the camshaft. It follows that, as the rocker arm swings from the straight-ahead position towards either lock, the peg axis travels through an arc that successively approaches, intersects and diverges from the vertical plane containing the shaft axis. This lateral movement results in a continuous variation in the required depth of meshing of the peg. Also, since the peg axis is only radial, relative to the camshaft, at two points on the arc of travel, the included angle of the track flanks varies with the amount of offset.

In the example already mentioned, the numerical reduction in the ratio towards each lock is 15·6 : 1 at the straight-ahead position, falling to 9·56 : 1 at full lock. This variation is, of course, obtained by varying the pitch angle of the track. It will be appreciated, however, that the lateral movement of the peg affects this angle also: towards one lock it is in the same direction as the rotation of the shaft, whereas towards the other lock it is in opposition. In the former case, therefore, the pitch angle should be greater than in the latter to give the same effective ratio on each side of the neutral position. Because of these various factors, the true form of the track is a complex one, and it is produced by a generating process, using a form tool of the same section as the peg: the tool follows the movement of the rocker shaft while the camshaft is appropriately rotated.

A single forging of En.16T steel comprises the rocker arm and its shaft. The shaft is slightly cranked to clear the piston sleeve, and is carried in two phosphor-bronze plain bearings. Of these, the lower is a Vandervell wrapped, steel-backed component, and is fitted in a long boss projecting from the main housing. Below it is a synthetic rubber seal of X-section, retained by a washer and circlip. The upper bearing, a solid bush, is in the rocker arm cover; its oil seal, also of X-section, is housed in a groove at the upper end of the shaft. Both bearings are lubricated by the hydraulic oil.

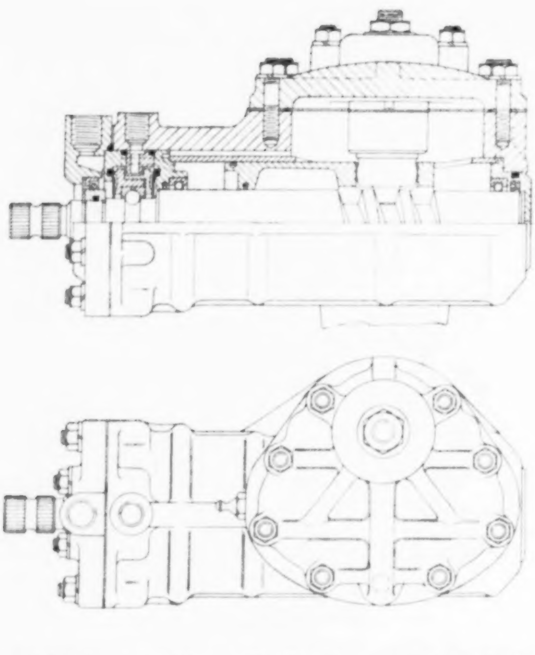
An unusual method has been adopted for supporting the rocker shaft, resisting cam thrust and adjusting the mesh of the peg: in effect, the shaft is suspended from the cover by means of the adjusting screw. Bored in the upper end of the rocker shaft is an axial, tapped blind hole. At the bottom of this hole is a smaller hole, into which is pressed

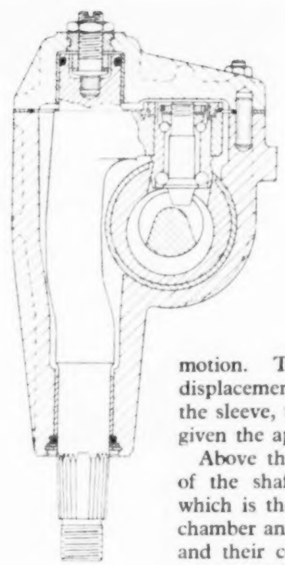
In the end covers, p.t.f.e. seals of castellated section are employed; the bottom cover carries the outer race of one of the camshaft bearings



a hardened plug. The adjusting screw has a button head at the lower end, and over its shank are slipped a Belleville washer and an externally threaded sleeve. This sleeve is screwed into the rocker shaft until the button is nipped between the plug and the washer. Then the cover is screwed on, and the assembly is offered up to the body.

After the nuts securing the cover have been tightened, the depth of engagement of the peg is set, by means of the adjusting screw, to give the minimum of backlash at the





General arrangement of the Hydrosteer gear; the seals now used between the end covers and the camshaft are of a later type. The method of adjusting the depth of meshing of the peg, and of taking the thrust from the cam, is unusual. A vertical dowel peg prevents rotation of the piston sleeve in the casing

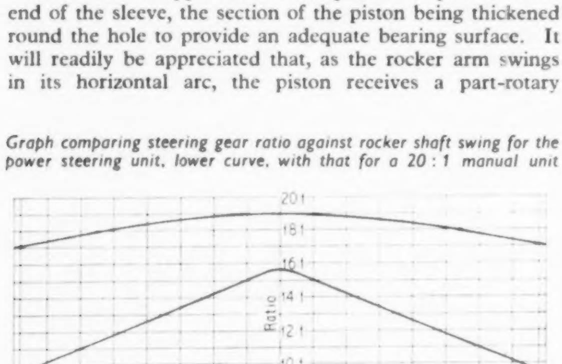
straight-ahead position. Finally, the screw is locked in position by an external nut. Dowels position the cover, which is secured by studs and a socket screw, all of $\frac{5}{16}$ in diameter. A metal gasket is fitted between the cover and the body, and it embraces a synthetic rubber sealing ring of square section.

The cam follower peg is made from En.34 steel, case-hardened, and its shape is a truncated taper of 30 deg included angle. Friction is minimized by permitting the peg to rotate, in two ball bearings of the cup-and-cone type. The two outer races are formed on a hardened, En.351 steel sleeve pressed into the rocker arm, against a shoulder, while the inner race of the lower, and larger, bearing is machined on the spindle of the peg. Because of the need for adjustment of these bearings, the cone of the upper bearing is a light push fit on the upper end of the spindle. A castellated adjusting nut is fitted above the cup, and is locked by a special cup washer, which is peened into two of the castellations. The sleeve that houses the bearings is secured in the rocker arm by another castellated nut and locking washer. Ten $\frac{1}{4}$ in diameter balls are used in the lower bearing and thirteen $\frac{3}{16}$ in diameter balls in the other.

A hole in the upper wall of the piston accepts the lower end of the sleeve, the section of the piston being thickened round the hole to provide an adequate bearing surface. It will readily be appreciated that, as the rocker arm swings in its horizontal arc, the piston receives a part-rotary motion. To accommodate the angular displacement of the axes of the hole and the sleeve, the lower end of the sleeve is given the appropriate spherical form.

Above the piston, near the upper end of the shaft, is the valve chamber, in which is the spool type valve. Both the chamber and the spool are of En.1a steel, and their combined design is such that, for assembly purposes, divided construction has to be employed for each: the chamber is made in three parts, located relative to one another by dowels, while the spool comprises two parts, screwed together. The chamber components are contained in the housing, and the spool is mounted on the shaft.

Graph comparing steering gear ratio against rocker shaft swing for the power steering unit, lower curve, with that for a 20 : 1 manual unit



The lower portion of the valve chamber houses the outer race for the upper of the two ball bearings carrying the camshaft. It is a push fit in the main casting and its lower end has a male spigot, which is pressed into the piston sleeve. In the bore, immediately above the race, is a groove containing an oil seal of similar type to that in the bore of the piston. On the upper face of this component is an annular groove, of stepped section, which houses the lower of the two valve seal assemblies. Each of these comprises an O-ring covered by a sealing plate—a thin steel annulus. The thickness of the O-ring is sufficient to hold the sealing plate just clear of its seating in the groove. Excessive spread of the ring is avoided by giving the bottom of the groove a 45 deg chamfer on the inner side. Encircled by the seal assembly is a shallow annular lip, which is the abutment for the lower of the two springs of the valve.

The intermediate portion of the valve chamber is of cylindrical form, with an inwardly projecting flange at its upper end. Machined in its periphery is an annular groove, which registers with the pressure connection from the pump. Four radial holes, equally spaced round the circumference, lead from the bottom of the groove to the inner face of the flange. The periphery is relieved at the lower end to accept the O-ring that forms the seal between the component and the body, in which it is a push fit.

Completing the chamber is the portion forming the upper end-wall. It has a short cylindrical extension that seats against the upper face of the radial flange on the intermediate portion. As are the other two components of the chamber, it is a push fit in the body, against which it is sealed by an O-ring in the same manner as the intermediate portion. This third portion carries the second valve seal assembly, and it also has an annular lip for the upper spring abutment.

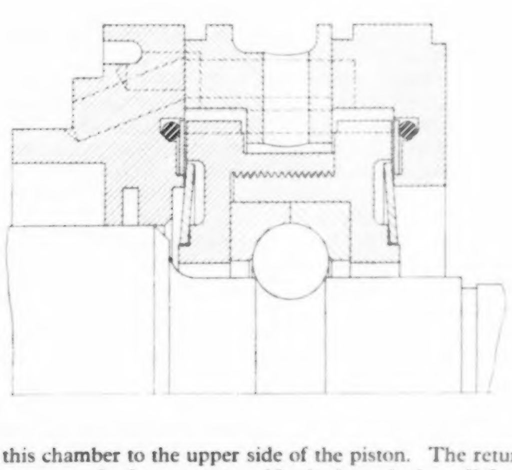
There is considerable radial clearance between the valve spool and the chamber, the insensitivity of the valve to foreign matter in the oil being attributable to this and to the end sealing arrangement. Mention has already been made of the divided construction of the spool. Between

the halves is clamped the transversely split outer race of a deep-groove ball bearing. This bearing positions the spool on the shaft while permitting the latter to revolve. Since the length of the spool is critical, the seating face between the halves is accurately machined on the end of the lower, female portion, and the race is shimmed as necessary for its correct fit in the spool.

The spool is of double-flanged form, and the inwardly projecting flange of the intermediate part of the chamber is within the waisted portion. In addition to the radial clearance between the inner face of the chamber flange and the waisted portion of the spool, there is axial clearance between that flange and the end flanges of the spool, the purpose of which will be explained later. On each end of the spool is an annular lip, of the same radius as the sealing rings in the chamber, and a spigot, which is of smaller diameter than the lip. On each spigot fits one of the washer type control springs; the spigots are relieved at intervals to permit the passage of oil past the springs. As is shown by the accompanying graph, the combined rate of the springs varies a little but is in the region of 15,800 lb/in. In the neutral position of the valve, the springs are preloaded in opposition, between the lips on the chamber end walls, the deflection of each being about 0·01 in.

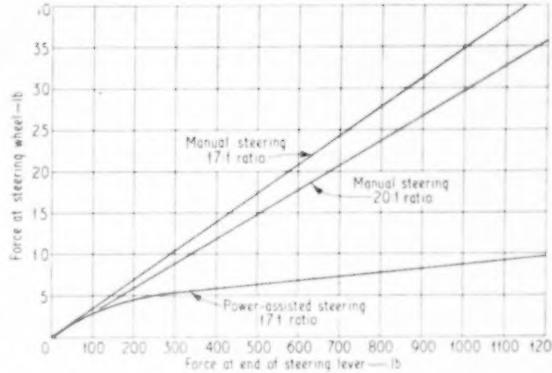
It can be seen from the appropriate illustration that there are four notches in the periphery of each of the two end flanges of the spool. In these notches sit steel rods, which pass through axial holes in the flange of the intermediate portion of the valve chamber. These rods fulfil the triple purpose of locking the halves of the spool together, of preventing the spool from rotating relative to the chamber, and of retaining the sealing plates in their grooves.

Some explanation is desirable here regarding the disposition of the various oilways. It has already been stated that the feed is to the intermediate portion of the valve chamber. Communication between the lower end of the chamber and the space below the piston is through an oblique drilling in the chamber wall to an axial groove cored in the body immediately above the piston sleeve. Passages formed in the three portions of the chamber lead from the upper end



of this chamber to the upper side of the piston. The return passage, on the low-pressure side, is through the reliefs in the valve spring spigots, and also through the clearances between the spool and the shaft, and between the shaft and the upper wall of the valve chamber.

A counterbore in the top cover fits closely over the upper portion of the valve chamber, so that the tightening of the cover clamps the complete internal assembly. Sealing between the cover and the body is effected by a square-section, synthetic rubber ring, and the cover is held down



In this graph, force at the steering wheel is plotted against force at the steering lever for typical manual units and for the powered system having a ratio of 17 : 1. The reduction in the effort is considerable

by eight $\frac{5}{16}$ in diameter studs and nuts. Since the cover is in contact only with the return side of the fluid circuit, high-pressure seals are not necessary between it and the shaft. However, the sealing arrangement is of interest since it makes possible the employment of the same size of seal as at the lower end, in spite of the smaller diameter of the shaft. The advantage here is that the longitudinal forces acting on the shaft are balanced, so there is no resultant load tending to move it in one direction or the other.

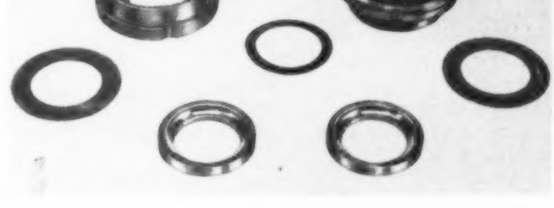
On the shaft, a short steel sleeve is held against a shoulder by a circlip fitting into a groove. It is a push fit on the shaft and its bore is grooved to take an O-ring. A second, smaller counterbore in the top cover houses a castellated p.t.f.e. seal identical with that at the other end of the shaft. This seal is retained by a washer and circlip, and it bears on the periphery of the sleeve, which is chromium plated to resist wear.

Principle of operation

Oil is supplied to the unit under pressure by a Hobourn-Eaton pump, of the latest type, in which hollow rollers are employed in place of vanes; this pump is belt driven from the engine. The hydraulic fluid, which is normally an engine oil of SAE 10W rating, is housed in a reservoir, of 2 pints capacity, which may be part of the pump assembly or be separately mounted. Flexible hoses connect the pump and reservoir to the steering box, and the couplings of the pump hose are of the high-pressure type.

From the valve chamber there are three possible paths for the oil: one is to the space above the piston, the second communicates with the lower side of the piston, while the third leads to the return to the oil reservoir. In the neutral

Left: Enlarged section of the valve and chamber assembly, showing the end seals, the preloading of the springs, and the divided construction of the valve and the bearing race. Below: The valve components; on the end of each half of the spool is a spigot that carries a spring

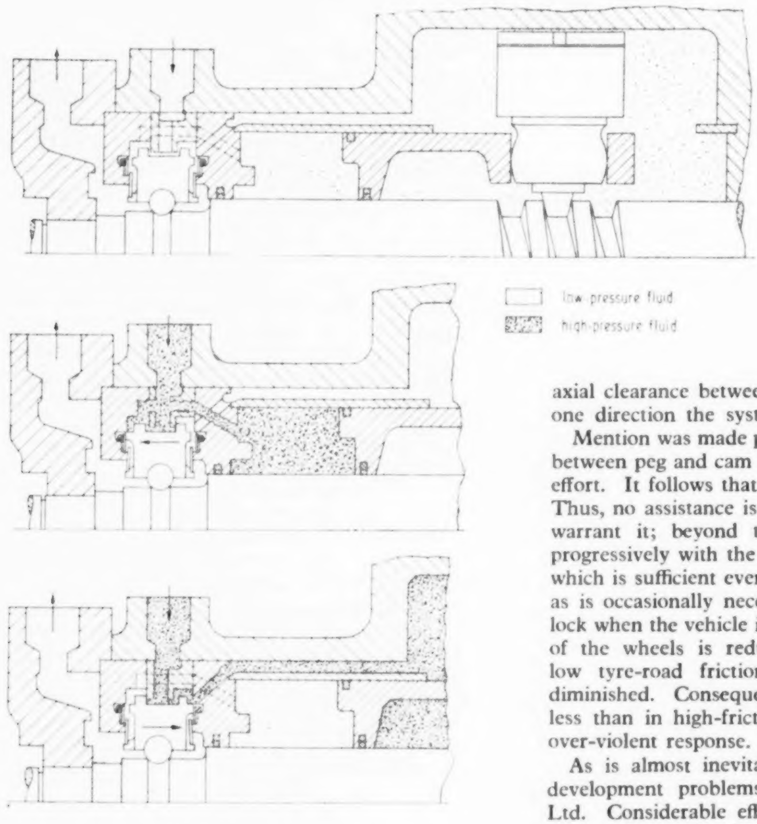


position, when no steering effort is being applied, the spool occupies a central position in the valve chamber. Neither ring seal is then in use, and fluid is free to circulate round the entire system, without resistance, and to return to the reservoir. This situation is illustrated in the first of the three accompanying operational diagrams.

Rotation of the steering wheel, and hence of the cam track, in either direction produces an axial reaction between the peg and the track. The magnitude of this reaction depends, of course, on the effort applied to the steering wheel and on the resistance to swivelling offered by the front wheels. Since the shaft has a limited amount of axial freedom, by virtue of the spool end clearances mentioned earlier, this reaction tends to cause the shaft and spool to move against the resistance of one of the springs.

If the reaction is strong enough, the load on the spool overcomes this resistance, and axial travel of the shaft occurs. This travel brings the lip on the appropriate end of the spool into contact with the adjacent seal, thereby closing the direct communication, at that end of the spool, between the oil supply and return passages. Direction of the oil pressure to the correct side of the piston is effected by introducing a flow restriction into one of the two remaining paths open to the oil.

The space between the end flanges of the spool and the flange in the chamber forms the communication between the two internal passages to the piston. It will therefore be apparent that, when the spool moves, and one of its flanges approaches the chamber flange, the entrance to one passage is restricted while that to the other is enlarged. The resulting pressure differential causes the piston to be moved in the required direction. Full powered assistance is attained on contact between the two approaching flanges, since the bleed from the high-pressure to the low-pressure side of the spool is then cut off. In the second and third operational diagrams, the conditions of maximum power assistance in the two directions are shown. Once the axial clearance between spool and chamber is taken up in one direction the system becomes mechanically positive.



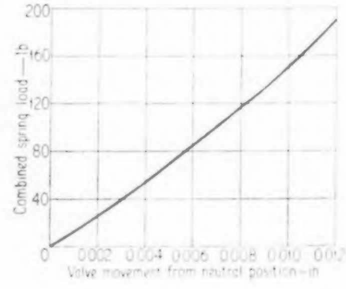
Diagrams indicating the passages for low- and high-pressure fluid within the Hydrosteer unit. In the top view no steering effort is being applied; in the second the piston is being moved to the right, and in the third its travel is reversed

Mention was made previously of the fact that the reaction between peg and cam track varies with the applied steering effort. It follows that the spool travel must vary similarly. Thus, no assistance is given until the effort is sufficient to warrant it; beyond that point, the assistance increases progressively with the effort up to the maximum available, which is sufficient even for violent manoeuvres at speed or, as is occasionally necessary, for putting the wheels on to lock when the vehicle is at rest. If the swivelling resistance of the wheels is reduced, however, as in conditions of low tyre-road friction, the reaction is correspondingly diminished. Consequently, the degree of power applied is less than in high-friction conditions, thereby avoiding an over-violent response.

As is almost inevitable with any new product, certain development problems were encountered by Hydrosteer Ltd. Considerable effort had to be devoted to obviating any stickiness in the action of the valve seals, and to eliminating oil leakage. Another difficulty experienced was the tendency of a pocket of air to form at the highest point

Below: The rocker arm and its cranked shaft are a single forging. On the end of the sleeve that carries the peg is a spherical seating face

Graph of the combined rate of the two valve springs, which work in opposition and are given a degree of preloading



of the rocker shaft housing. Consequently, a bleed screw is now fitted at this point, in the cover, and bleeding is carried out in the same manner as on hydraulic braking systems. A more recent refinement is the embodiment of a hole in the piston wall, which at full lock permits a bleed to take place from the high-pressure to the low-pressure side, thus avoiding hydraulic overloading of the stops.

The design objective in respect of road behaviour was referred to initially. Its attainment was recently confirmed in a test run on a much used Super Snipe car. The route taken embraced town running, main roads and lanes, and included several acute corners, that had to be taken at low speeds, and more than one series of fast bends involving rapid changes of lock. Only in respect of the lightness of the steering was there any obvious indication of power assistance, and the feed-back was sufficient to avoid "deadness". The gearing finally chosen for this installation would be difficult to improve upon: only about 2¾ turns of the steering wheel were needed from lock to lock and, though control was positive—without being excessively sensitive—at high speeds, no substantial effort was needed near full lock when the car was crawling, or even when it was at rest. It was considered that low-speed manoeuvring involved no more work than it would with the average 1-litre car weighing about 15 cwt. The self-centring effect was mild, but adequate in view of the relatively high gearing.

Trace Reading Equipment

A NEW means of rapidly and accurately measuring oscillograph traces is now available. It is the TRE12 trace reading equipment manufactured by Southern Instruments Ltd., of Camberley, Surrey. The measured ordinates are automatically corrected for recording system non-linearities, and scaled to present a digital output in the original units. This output is displayed visually, and can also be tabulated by a typewriter, punched on tape or cards or any combination of these. All the operating controls are convenient to handle, and are so placed as to reduce operator fatigue to a minimum.

The equipment consists of a trace reader and a 4 ft rack containing power supplies, digitizer, scaling amplifier, range setting facilities and output units for feeding the typewriter, tape perforator, etc. The reader consists of a well styled casing enclosing the paper drive mechanism, which is contained in a skeleton drum mounted with its axis horizontal. To load the record spools, the whole drum can be easily rotated to the required positions. Paper up to 12 in wide is motored across the reading platen by push-button controlled magnetic clutch drives. A two-speed gearbox is similarly operated, providing paper speeds of 1¼ in/sec and 12 in/sec. When neither speed is in use the paper is held taut and can be moved by a hand wheel.

To the operator's left is a double-ended cursor, which is moved by a spin wheel. The upper end rests on the record while the lower rests on a 3 in diameter function drum mounted horizontally at the front of the reader. This drum can be rotated by a manual control on the right. To it can be attached a sheet of drafting card accurately registered in position. By replacing the detachable lower cursor by one suitably pierced, a calibration curve is drawn on this card. For the reading of records, the cursor is moved horizontally to the required point, and the function drum is then rotated until the calibration curve is under the lower cursor. The horizontal cursor movement is thus converted into a rotation of the function drum and, by means of a potentiometer attached to the drum, a voltage is developed in proportion to the rotation.

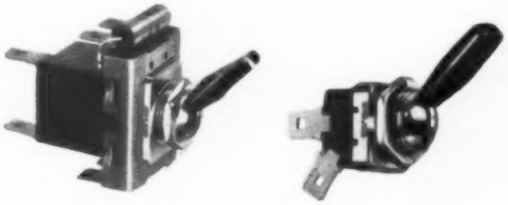
Operation of a read-out push-button or foot switch brings into action the digitizer, which converts the output voltage from the reader into digital form, displays it visually as illuminated figures, and also provides signals to operate the typewriter or other output organ. A programming panel is provided to enable the required digitizer scale to be selected. Twelve switches allow scales to be set up for any number of channels up to twelve in any desired order. A suitable signal is returned to gas-discharge numerical indicators on the trace reader showing the next channel to be read.

Other principal features of the equipment are as follows. It covers ranges of ± 999, ± 1999, ± 4999, can be calibrated linearly or non-linearly, and is accurate to $\pm 0{\cdot}1$ per cent of full scale. Trace records can be on either opaque film or paper, perforated or unperforated, and can be in photographic or direct writing, continuous or intermittent. The spools take paper up to 12 in wide, and hold 5 in on a 1 in diameter core. Print-out can be effected in 1·25 sec, and full scale output can be obtained for any trace up to 12 in amplitude. The mains supply voltage required is 200 to 250 volts, 50 c/sec a.c., and the consumption is approximately 750 watts. So far as dimensions are concerned, the trace reader is $26\frac{1}{2} \times 25 \times 16$ in high, and the console has overall base dimensions of 2 ft 6 in square and its overall height is 4 ft 10½ in.

The use of drafting card for calibration curves, with its resultant stability and ease of exact replacement, has made the repeatability of measurements a most useful feature. It enables records and calibration curves to be stored and remeasured at a later date without loss of accuracy. This instrument is already in use in the aircraft industry, where it is used for analysing records obtained on flight tests.

Lucas Lever Type Switches

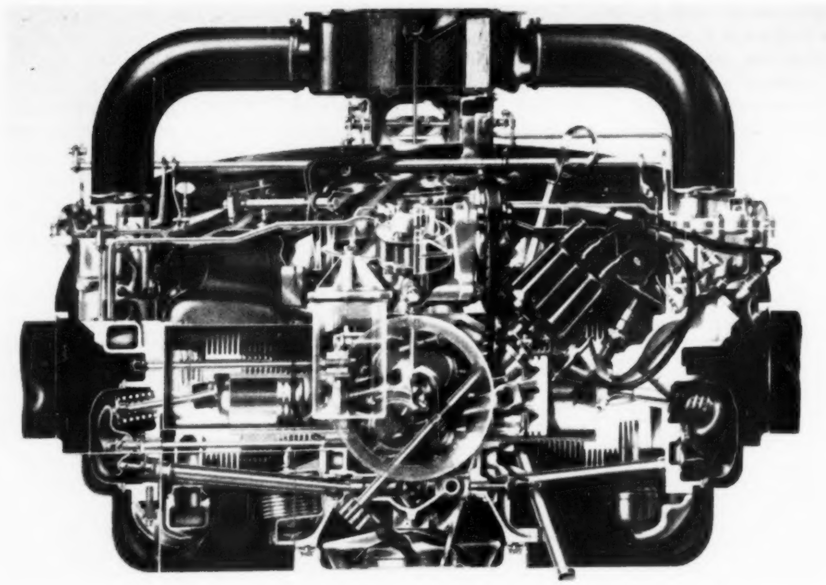
SEVERAL of the latest British cars are fitted with lever type electric switches from the new Lucas range. The lever is smartly styled, and the toggle mechanism is robustly designed to ensure reliable operation over long periods. In the range are several different two-position and three-



Typical examples from the new range of Lucas toggle-action switches: the three-position model 57SA, left, and the two-position model 65SA

position switches, all fitted with Lucar blade connectors for the wiring.

An example of a three-position switch is illustrated. It is the model 57SA, which is intended for flashing type direction indicators and is fitted with a dimming resistor for the warning light. The other switch depicted is the least expensive in the range, the model 65SA; this is a two-position type suitable for fog lamp operation, though it can be obtained with a spring return action for use in conjunction with screen washing equipment. These switches are now in quantity production by Joseph Lucas Ltd., the address of whom is Great King Street, Birmingham, 19.



An interesting feature of the induction system of this engine is the single automatic choke installed in the air intake tube to the central oil-wetted cleaner

CHEVROLET AIR-COOLED ENGINE

The Six-Cylinder, Air-Cooled Engine Designed for the General Motors Chevrolet Corvair

WITH a six-cylinder, light-weight, air-cooled engine mounted at the rear, the new Chevrolet Corvair marks a completely new departure in American motor-car design. This layout has certain advantages in respect of low floor and even some aspects relative to weight distribution. Nevertheless, the engine must be light and compact, especially in the fore-and-aft dimension, if road handling characteristics are to be satisfactory. For these reasons, the new Chevrolet engine is air-cooled, is largely made of aluminium and has its six cylinders horizontally opposed. Since this engine has been designed by a manufacturer who certainly understands the needs of mass production, it is worthy of serious study.

Dimensions and output

The bore and stroke are $3{\cdot}375 \times 2{\cdot}6$ in, and this gives a swept volume of 140 in^3, or 2·3 litres. A compression ratio of 8·0 : 1 has been adopted and the engine is intended to be run on regular-grade petrol. The maximum power output of the unit, as installed, is 80 b.h.p. at 4,400 r.p.m., which corresponds to a mean piston speed of only 1,900 ft/min, and the maximum torque is 125 lb-ft at 2,400 r.p.m; this corresponds to a b.m.e.p. of 136 lb/in^2 If the cylinders are numbered, conventionally, from the front of the engine, that is, the end opposite to the flywheel —in this installation the end nearest the rear of the car— the left bank of cylinders includes numbers 1, 3 and 5, while the right bank contains numbers 2, 4 and 6. The firing order is then 1-4-5-2-3-6 and, again as viewed from the front end, the direction of rotation of the engine is anticlockwise.

Piston and crankshaft

A combustion chamber of a slightly-tilted bath tub form is employed, while the piston crowns are flat. The tin plated, cast aluminium alloy pistons are of the slipper type, with steel struts to control expansion. Each piston has two $\frac{5}{64}$ in wide compression rings, with coated rubbing surfaces, and one $\frac{3}{16}$ in oil-control ring. A weight of 14·6 oz is quoted for the bare piston. The gudgeon pin is 0·80 in diameter and is secured by pressing it into the connecting rod.

The connecting rods are of conventional two-bolt design and are of forged steel. Their centre-to-centre length is 4·72 in, and the bare weight of each is 13·73 oz. Moraine Products Division supply the big-end bearings and the main bearings, which are steel-backed, with a sintered copper-nickel matrix and a thin lead-base white-metal overlay.

There are six 1·800 in diameter separate crankpins and four 2·0983 in diameter main journals on the forged steel crankshaft. The crankpins are arranged in pairs, each pair being set mutually at 180 deg, in a plane at 60 deg to that of the next. With this arrangement of crankpins in pairs, without main bearings between, it has been decided that counterweights are not necessary.

Crankcase and cylinders

The crankcase is of cast aluminium alloy and is, in effect, made up of two conventional three-cylinder crankcases bolted together at their sump faces. There are no main-bearing caps, the bearings being clamped between the halves of the crankcase. Nor are there any separate camshaft bearings, since the camshaft runs directly in the aluminium crankcase. A rectangular, cast aluminium cover is bolted to the top of the crankcase assembly and a shallow sheet-steel sump is bolted to the bottom. The front cover of the engine, again designated in the conventional sense— it is, in fact, toward the rear of the car — contains the distributor and oil pump drive, and incorporates mounting faces for the fuel pump and oil filter. Each half of the crankcase has three machined openings, into which the lower ends of the cylinder barrels fit.

Both ends of the individually cast, iron cylinders are open, and 14 fins are, of course, integral. These cylinders are

mounted 4·85 in apart on the crankcase, and are secured by long studs which pass through the cylinder head. The two aluminium alloy cylinder head castings are held in place by four long, waisted studs per cylinder, or 12 studs per head. Two of these studs on each cylinder have extensions to secure the self-aligning rocker arms. Each head has seven continuous, all-round fins and there are eight additional fins at each exhaust port. A short steel tube is pressed into each exhaust port and extends downward into the cast iron exhaust manifold, into which it is sealed by a triangular-section ring. Each exhaust manifold is secured to the cylinder head by three sheet-steel clamps. The single exhaust silencer is mounted on the engine. Cast nickel-steel valve seat inserts are pressed into the intake ports, while cast chromium steel inserts are pressed into the exhaust ports. The 14 mm sparking plugs are mounted on the upper side of the head, along the top outer edge of which is cored the intake manifold.

Valve gear and fuel system

The valves are conventional in design and have 45 deg seats. They are actuated by means of self-adjusting hydraulic tappets, hollow steel push rods with spun-over spherical ends and pressed steel rocker arms. No rocker shaft is used and a spherical spacer and self-locking nut are provided with which to set the initial position of the self-adjusting tappet pistons. Oil is fed through drilled galleries in the crankcase and thence through the tappets and push rods to the rocker arms. It is returned to the sump through the tubes that house the push rods.

Four bearings carry the cast iron camshaft, which is driven by a pair of gears, that on the crankshaft being of steel and meshing with a cast aluminium gear on the camshaft. While the inlet cams are conventional, the exhaust cams have to be placed very close together so only three are used, each of double width. The valve timing is: inlet opens 15 deg B.T.D.C. and closes 37 deg A.B.D.C; exhaust opens 59 deg B.B.D.C. to 13 deg A.T.D.C; the overlap is thus 28 deg.

Each of the two inlet manifolds is served by a single Rochester Products, 1¼ in carburettor, specially developed for this engine. Air, drawn through one oil-wetted air cleaner, reaches the carburettors through two reinforced rubber hoses. An interesting feature is that the single automatic choke is placed in the short tube through which air goes to the air cleaner. The mechanical fuel pump is mounted on the front cover of the engine and is driven by a push rod and eccentric on the crankshaft. Thus, it is driven at twice the usual speed and, for this reason, has a relatively short stroke. To avoid vapour lock, the pump is mounted outside the engine cooling duct.

Lubrication and cooling

The gear type oil pump is conventional in design, except in that its body is a part of the aluminium front cover. It is driven by the shaft that also drives the distributor. Oil is drawn from the 3½ quart sump through a large fixed screen and is pumped through passages, formed between the crankcase and the front cover, to the full-flow oil filter—a by-pass valve permits oil to flow past the filter, should the element be clogged. If the oil is warm it then flows through an oil cooler in the cooling air duct, otherwise the oil cooler is by-passed by a thermostat valve until the temperature reaches 160 deg F.

From the cooler, the lubricant flows through two drilled galleries in the crankcase to the tappets and valve gear and, through other drilled passages, to the main and camshaft bearings. The connecting rod big-end bearings are served through holes drilled in the crankshaft. For ventilation, the crankcase is fitted with a simple oil separator and a tube which extends downward from the crankcase. Apart from this tube, the crankcase is sealed, since it is felt that, with the rapid warm-up of an air-cooled engine, fresh-air ventilation is not needed to prevent condensation.

Cooling air is drawn in through an 11 in diameter centrifugal blower, mounted with its axis vertical above the centre of the engine. This arrangement affords excellent distribution of the air with simple ducting. The impeller is driven by means of a V-belt, at 1·58 times engine speed, and delivers 1,800 ft^3/min of air at an engine speed of 4,000 r.p.m. Above the driving pulley on the front end of the crankshaft, the two strands of the belt are turned through a right angle by two pulleys: one is on the 12-volt dynamo, which is mounted more or less at right angles to the crankshaft, and the other is an adjustable idler. Since the blower and the dynamo thus have a common drive, failure of the belt will be indicated on the ammeter.

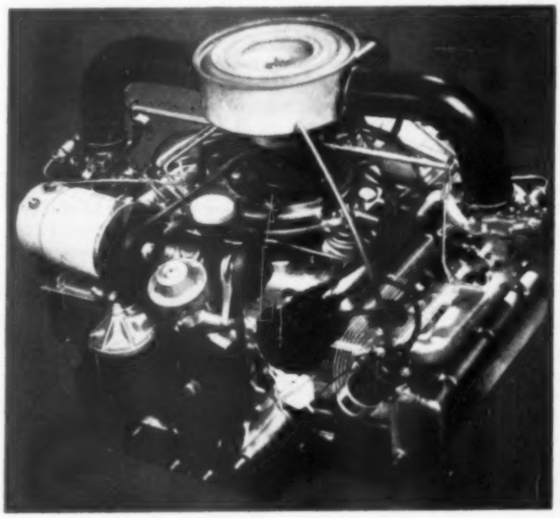
The blower enclosure forms part of a sheet-metal baffle that covers all of the engine out as far as the intake manifolds. Having passed the cylinder barrels and cylinder heads, the warm air is collected in ducts on each side of the sump and is discharged toward the rear of the car. All openings between the engine and the body are carefully sealed by a horizontal panel, to prevent recirculation of the warmed air. However, this seal is arranged so that the body can be lifted off the engine-gearbox-axle assembly without unfastening any sealing parts. An interesting feature of the cooling system is the provision of a thermostatically-controlled air baffle in the blower intake. This baffle is shaped rather like a wheel rim, 7½ in diameter by 2½ in wide, which is lowered towards the eye of the impeller until the temperature of the air in the discharge ducts reaches 180 deg F. When this temperature is attained, the baffle is raised again by a Harrison bellows type thermostat.

Considerable effort has been made to keep the engine quiet, by the use of a centrifugal blower, self-adjusting tappets, closely fitted, controlled-expansion pistons and so on. In addition, the panel between the passenger and engine compartments is insulated with a 1 in thick glass-fibre sheet on the side adjacent to the engine and a ⅜ in jute cover on the other side.

Comparison with the other compact cars

It is of interest to compare, very briefly, the Chevrolet Corvair with the other two new American compact cars, the Chrysler Valiant and the Ford Falcon. All these cars are 70 in wide, approximately 54 in high, and from 180 to

View showing the layout of the induction and air-cooling systems



184 in in overall length. While the Corvair and the Falcon are said to weigh 2,275 lb, the weight of the Valiant is about 2,600 lb. The Corvair is said to carry 40 per cent of its weight on the front and 60 per cent on the rear wheels, and it is claimed that this weight distribution does not vary greatly whether the vehicle is empty or laden.

In comparison with the engine we have just described, the other two cars have power units that are thoroughly conventional. Both have water-cooled, six-cylinder in-line engines rated at 90 b.h.p. That of the Falcon is similar in layout to the British Ford six-cylinder engine; a point of interest is that the intake manifold, while outside the cylinder head water jacket, is cored in the head. The Valiant engine is conventional, but is tilted at 30 deg towards its right-hand side in the chassis. It is fitted with a spider type intake manifold, with very long branches, and has an a.c. dynamo.

Closely associated with the Corvair engine are the gearbox and differential. Two types of gearbox are offered: one is a manually-operated, three-speed unit with ratios of 3·22 : 1, 1·84 : 1 and direct-drive in third, and the second is the Powerglide automatic transmission consisting of a three-element torque converter driving a two-speed planetary gearbox, which has an overall maximum gear ratio of 4·73 : 1. With either system, the axle ratio may be either 3·55 : 1 or 3·89 : 1.

All in all, the Chevrolet Corvair, with its single unit body and frame and its light-weight, air-cooled engine mounted at the rear, represents a complete break-away from the somewhat hidebound American motor-car tradition. It will be interesting to see how it performs in owners' hands and to what extent it and its two American competitors will stem the foreign invasion of the American motor car market.

Translations of German Standards

THE following is a list of German standards translated into English and published in February 1960:

DIN UDC

621-229 Tool and work holders—6319, HOLDING FIXTURES: *spherical washers, conical seats, clamping bolt heads, conical countersinks, July* 1944*X*

621-5 Regulation, servicing—950, HANDWHEELS: *offset-arm type, hub bored round, March* 1959

621-72 : 629.11 Lubrication devices for land and road vehicles—71412, AUTOMOTIVE ENGINEERING: *lubricating nipples, September* 1950*X*

621.3.5.626 Bushings—42531, POWER TRANSFORMERS: *bushings for indoor and outdoor use, voltage rating from* 10 *to* 30 *kV*, 200 *A*, *July* 1959

621.316.541 Plug-and-socket devices—41622, JACK PANELS WITH BLADE CONTACTS, SMALL TYPE: *principal dimensions, June* 1959

621.643.2 Pipes, conduits—2448, SEAMLESS MILD STEEL PIPES FOR PIPELINES AND CONSTRUCTION, JANUARY 1940*X*

621.78 Heat treatment of metals, furnaces—17014 HEAT TREATMENT OF IRON AND STEEL: *technical terms, October* 1959

621.822 Bearings—5402, ROLLING ELEMENTS: *Sheet* 1, *cylindrical rollers, short rollers, August* 1942; *Sheet* 2, *long rollers, August* 1942; *Sheet* 3, *for anti-friction bearings; needles, February* 1957

621.824 Shafts—5472, MACHINE TOOLS: *spline shafts and spline bore profiles with* 6 *keys, internal centring, May* 1952

621.882.082 Screw threads—259, WHITWORTH PIPE THREAD: *Sheet* 1, *parallel internal and parallel external thread, nominal sizes, May* 1959; *Sheet* 2, *tolerances, preliminary standard, May* 1959; *Sheet* 3, *limits, preliminary standard, May* 1959

667.6 Paints and varnishes—55928, PROTECTIVE COATINGS FOR STEEL STRUCTURES: *rules, June* 1959

667.6 : 620.1 Testing of paints and varnishes—53213, TESTING OF MATERIALS FOR SURFACE COATINGS AND SIMILAR PRODUCTS CONTAINING SOLVENTS: *determination of the flash point in the closed crucible, February* 1958

669 Metallurgy—1353, SYMBOLS FOR SECTIONS, BARS, SHEET AND PLATE, STRIP AND SIMILAR MATERIAL, SEPTEMBER 1959

669.13/.15 Cast iron, steel—1651, FREE CUTTING STEELS, QUALITY STANDARDS, AUGUST 1954. 1691, GREY CAST IRON, UNALLOYED AND LOW-ALLOYED, NOVEMBER 1949*X*

669.14-41 Sheets, strips—1543, MILD STEEL ROLLED: *steel plate over* 4·75 *mm* (*heavy plate*), *dimension and weight variations, July* 1959. 41301 ELECTRO SHEETS: *magnetic materials for transformers, May* 1954

669.14-42 Sections, profiles—1025, STEEL SECTIONS: *Sheet* 1, *hot-rolled narrow I-beams, I-range, dimensions, weights, static properties, July* 1959; *Sheet* 2, *hot-rolled wide I-beams* (*I-section wide flange beams*), *I PB-range and I B-range, dimensions, weights, static properties, July* 1959; *Sheet* 3, *I PB*1*-range, light pattern, dimensions, weights, static properties, July* 1959; *Sheet* 4, *I PBv-range, heavy pattern, dimensions, weights, static properties, July* 1959; *Sheet* 5, *hot-rolled medium-width I-beams, I PE-range, dimensions, weights, static properties, July* 1959
1026, STEEL SECTIONS, BARS, *hot-rolled rounded-edge channels, dimensions, weights, static properties, July* 1959. 1027, STEEL SECTIONS, BARS: *hot-rolled rounded-edge Zeds, dimensions, weights, static properties, July* 1959. 1028, STEEL SECTIONS, BARS: *hot-rolled rounded-edge equal angles, dimensions, weights, static properties, July* 1959. 1029, STEEL SECTIONS, BARS: *hot-rolled rounded-edge unequal angles, dimensions, weights, static properties, July* 1959. 1612, MILD STEEL, ROLLED: *steel sections, bars and wide flats, March* 1943*X*. 59200, HOT-ROLLED WIDE FLATS: *dimensions, permissible variations, weights, September* 1959

669.2/.8 Non-ferrous metals—1725, ALUMINIUM ALLOYS: *Sheet* 1, *wrought alloys, May* 1958

677.052 Spinning, twisting, doubling machines and accessories—64045, RING SPINDLES WITH PLUGS FOR LIGHT AND HEAVY WARP TUBES ACCORDING TO DIN 64063, JULY 1950. 64063, LIGHT AND HEAVY TUBES FOR RING SPINDLES ACCORDING TO DIN 64045: *tube gauges, July* 1950

677.054 Warping, weaving machines, looms—64550, LOOMS: *designation of inter-related dimensions, standard widths from* 800 *to* 3,500 *mm, August* 1957

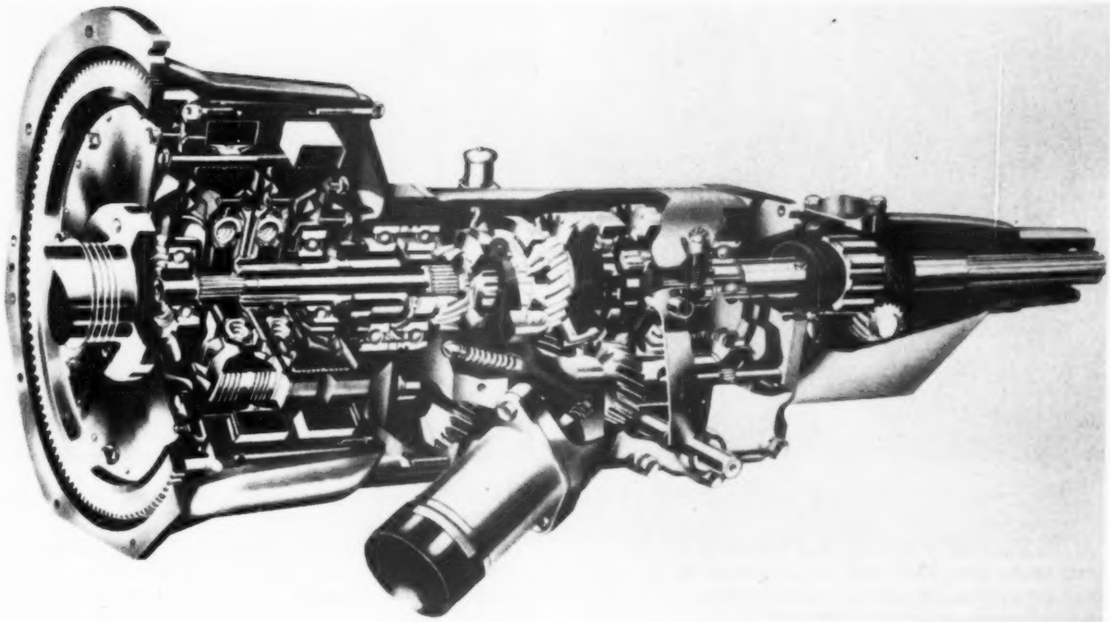
678.5/.8-4 : 620.1 Form and shape of products, testing—53358, TESTING ARTIFICIAL LEATHER: *determining the area weight of an artificial leather cloth coating, October* 1957. 53382, PLASTIC FOILS AND ARTIFICIAL LEATHER: *effect of flame test on one side, swivel burner method of testing, October* 1957

Other lists of translations, as they become available, can be obtained free of charge from the Beuth-Vertrieb GmbH, Berlin W15, Uhlandstr. 175, Germany. So far, 820 English translations have been made: they are prepared by a special department of the German Standards Committee.

Heavy-Duty Grease Pump

HEAVY lubricants and bituminous materials can be delivered under high pressure by the Hercules grease pump introduced by Centralube Ltd., Eley Estate, Edmonton, London, N.18. This pump is foot operated and can deliver at pressures of up to 15,000 lb/in²; a handwheel is screwed down every 20 to 40 applications, to ensure full charging of the pressure chamber. The output per stroke may vary slightly with the type of grease, but it is 2·3 gm in the case of a grease of penetration 250.

By virtue of its high-pressure delivery, the Hercules pump should be equally suitable for chassis and machine lubrication, particularly where long pipelines are involved. Since it can be used with positive dividers in the delivery line, it can be regarded as an intermediate stage between the individual type of grease gun and a centralized lubrication system. The pump is of compact dimensions, weighs just under 50 lb and is fitted with wheels for ready mobility.



SMITHS AUTOMATIC TRANSMISSION

Production of Units for the Hillman Minx "Easidrive" Model

THIS automatic transmission unit is unique by reason that its main components are two dry-particle, electromagnetic couplings. The principle of this type of coupling was first postulated, and demonstrated in the laboratory, by Jacob Rabinow in the U.S.A., and was subsequently patented. Original development work to utilize the principle in clutch units for commercial application was carried out by the Eaton Manufacturing Co. of Cleveland, Ohio. S. Smith and Sons (England) Ltd. later acquired from the Eaton Manufacturing Co. the exclusive patent rights for this type of coupling in all countries with the exception of North America. Over a number of years Smiths have conducted further basic research on the principle and have undertaken intensive development of practical units for industrial and, particularly, automobile application.

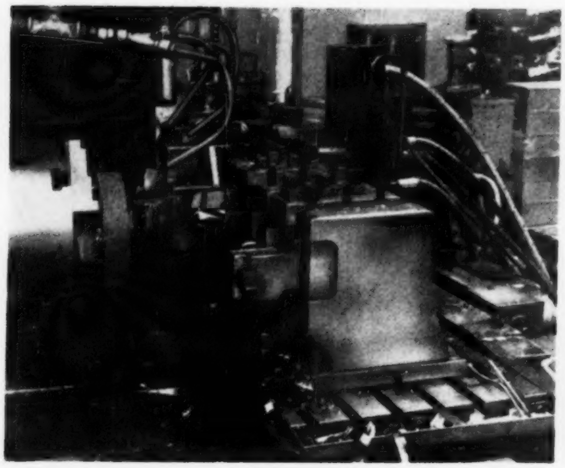
Many prototype installations of automatic and semi-automatic transmissions have been made for exhaustive testing and investigation, and a vast amount of operating experience has been accumulated in running many hundreds of thousands of miles on road and track. A description of the design and functioning of a typical Smiths automatic transmission has already been published*. All references and illustrations in the present article relate to the version specifically "tailored" for the Hillman Minx "Easidrive" model, currently available as an option. The car manufacturer supplies the mechanical three-speed gear unit and bell housing and at Smiths Witney factory, the magnetic coupling unit, together with a solenoid and wiring harness, is fitted to the gearbox. The assembly is then tested and passed out as a complete transmission. The other items making up the complete transmission system that are produced at Witney include the control unit, governor and selector switch.

*"*Automobile Engineer*," *September*, 1957.

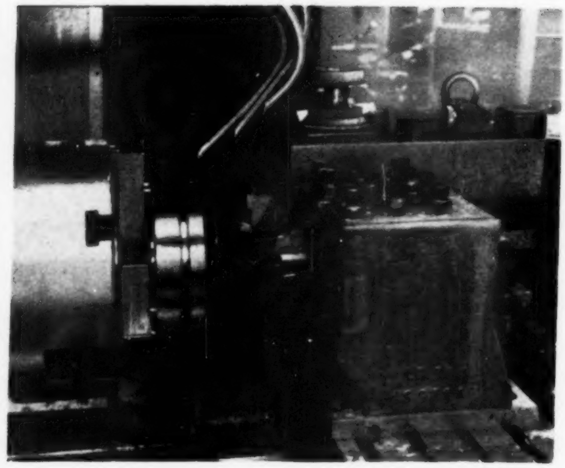
As might be expected, tooling, machining and assembly methods utilized in the production of such a novel mechanism differ markedly from those employed in the manufacture of more conventional units, and present a number of features of interest. The main rotating members—driving and driven elements—are held to close dimensional tolerances and a high standard of surface finish is maintained. Balance is of prime importance, both in the case of individual components and also collectively after assembly.

These parts are press forgings and drop forgings in En2A/1, having the low carbon content—maximum 0·15—specifically necessary to ensure a good magnetic permeability with low remanence. They are fully annealed and are cleaned by shot-blasting. Rough machining and finish machining is carried out on a battery of Müller PF350 multi-tool and copying, face-turning lathes, with 18/30 kW and 7/10 kW two-speed motors respectively. Each is equipped with the maker's "Eltropilot" programming and control system to give completely automatic operation. Speeds, feed rates, and all machine movements are planned and sequenced and then set up on a plugboard. For repeat settings a punched programme board can be overlaid on the plugboard to mask unwanted sockets.

Trips on the machine are carried on demountable plates and can be set up in the toolroom. These may be stored ready set-up for repeat operations. Final positioning when setting up the machine is by micrometer adjustment of the trip switches. Tooling also can be set up away from the machine and the multi-tool blocks finally adjusted for position on the T-slotted cross-slide table. Changeover of setting from one job to another is effected by a single machine setter in 3-4 hours for light tooling and 4-6 hours for heavy tooling, under all normal circumstances.



Müller multi-tool set up for roughing front driven member



Finishing rear driving member on Müller programmed multi-tool lathe

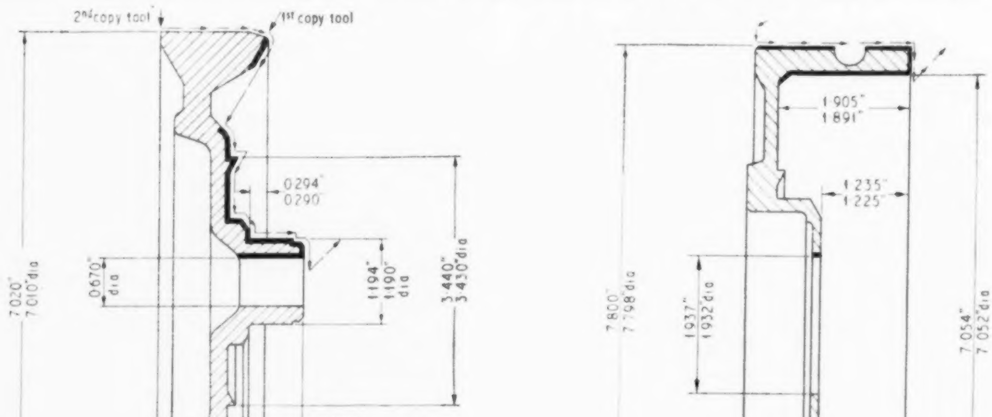
All longitudinal, transverse, and combined 45 deg movements of the cross-slide table are controlled by the "Eltropilot" system but for other, shaped surfaces on the workpiece a copying attachment mounted on the cross-slide is employed. In some instances—roughing the front driven member, for example—two copy tools are used. With the exception of the core drill, all cutting tools are carbide tipped and ground with a front angle of 10 deg and up to 20 deg top rake. Clamped tool bits are used wherever possible, to obviate loss of time in regrinding and setting. Mainly they are of the throwaway-bit type but some are of the band-clamped, regrindable-slug pattern. Band-clamping types are used specifically to facilitate setting-up of tools for copying operations where accurate replacement is essential. Sandvik Coromant grade S4 is chosen for roughing and S1P grade for finishing and copy turning. For some roughing operations, in which heavy cuts may be experienced, and also some finishing operations, Ringspan chucks are employed instead of jaw chucks.

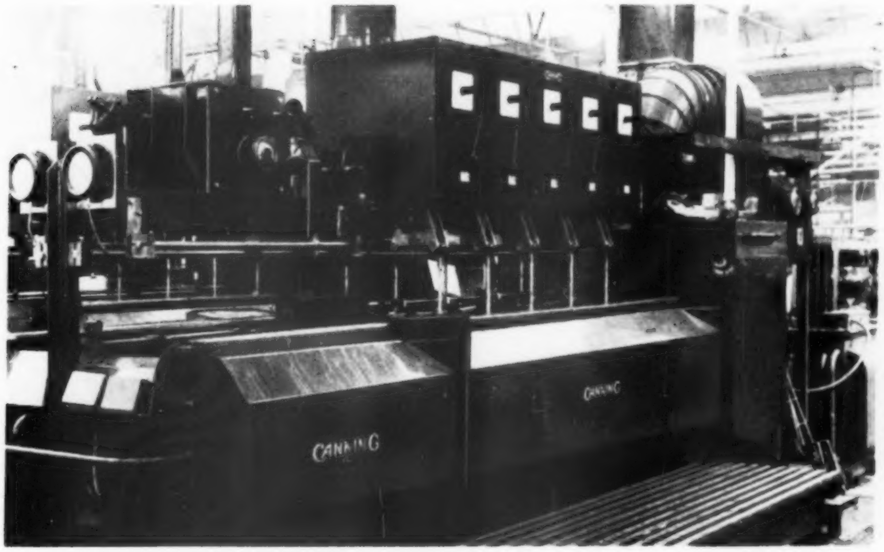
The rear driving member provides an example of finishing operations on a Müller machine. A multi-tool set-up is used to finish the outside diameter and the main bore to limits of 0·002 in, face the end square to 0·001 in, and chamfer the entry. The copying head used for finishing the outside diameter is provided with a means for independent adjustment on the tool slide.

Another ring-type workpiece, though not a rotating one, is the central member of the stator which is registered in and bolted to the bell housing. This is the largest and heaviest component to be finished on the Müller lathes, being approximately $10\frac{7}{8}$ in diameter × $3\frac{1}{16}$ in wide. It is held on the outside diameter in a three-jaw chuck, with the jaws extended by pads which serve as back stops. The operating speed is 405 rev/min giving linear speeds of 1,200 and 1,100 ft/min on external and internal registers respectively, and the feed rate is 0·006 in/rev.

The outer register and face is a copying operation, using a Sandvik Coromant, F1 grade, band-clamp tool. All

Machining operations on Müller lathes. Left: rough turning front driven member. Right: finish boring and turning rear driving member





Canning semi-automatic, multi-unit, hard chrome plating plant

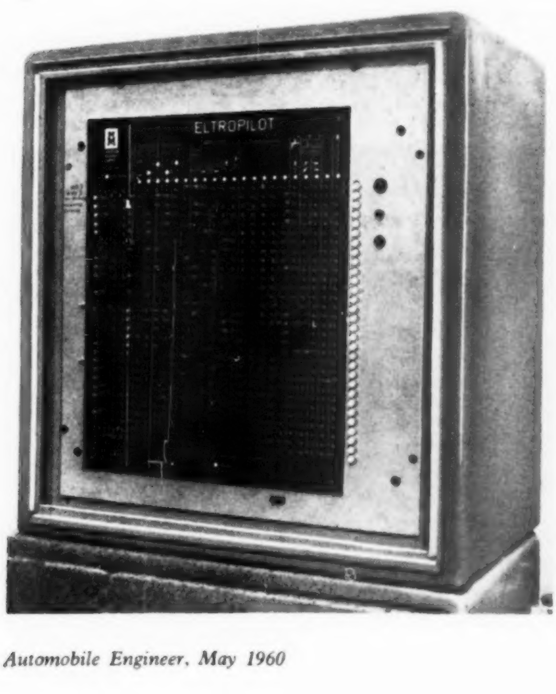
boring is done with a cluster-type bar fitted with Herbert Microbore tools and chamfer tools, to a total of six. Microbore tools are tipped with the new hard grade of A.C. carbide, ground with a 12 deg top rake.

In the stator and the driving and driven members numerous holes are required; some clearance and some tapped. Drilling is performed on a Corona 21A power-feed machine fitted with one of several Mulhead multiple drill head and work fixtures. The work is located on the base of the fixture with two pin registers and a radial position pin, and the drill plate, sliding on vertical columns, is spring pressed on the component as the drill head descends. A changeover of multi-head and location fixture for work on a different component can be effected in 60 to 90 minutes. Alongside each drilling machine is a Kerry single-spindle, sensitive feed drill which is used to deburr clearance holes and chamfer tapping holes. This work is performed by the drill operator during the multiple drilling cycle time.

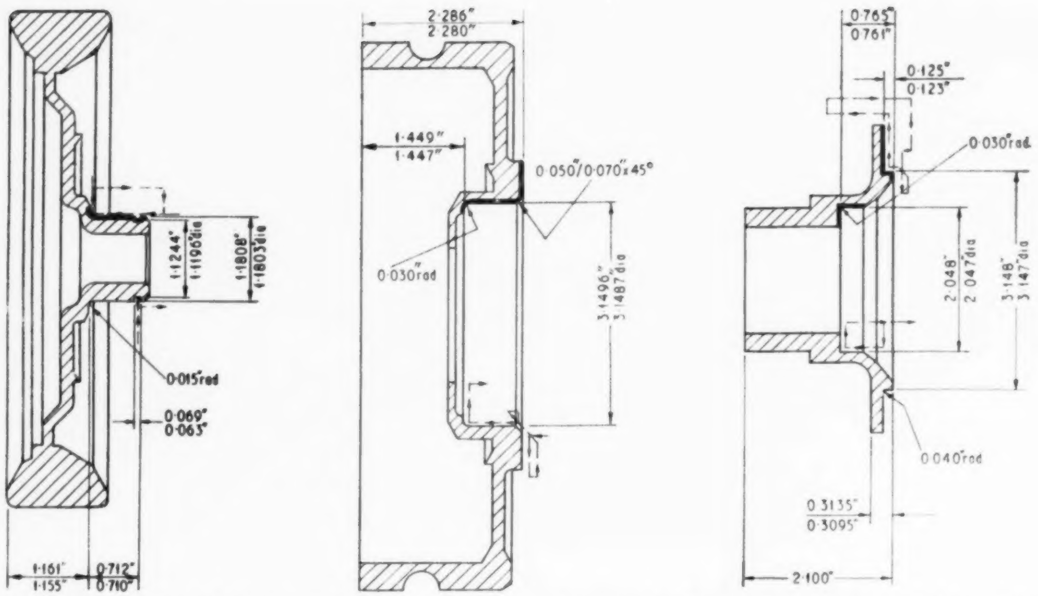
For tapping these holes, a Huller No. 5, pitch-control machine, with a motor auto-reversing on a trip switch, is equipped with a Mulhead multi-spindle head. The work table of the fixture is slid manually on guide rods to a stop, locked at the working position, and retained by a Speetog clamp. Location of the component on the table is by four pin registers and a radial position pin.

Precision boring and turning of the hub centres of rotating components is done on New Britain, two-spindle, fine-boring and turning machines. Components are held in three-jaw, air-operated chucks and each setting is checked for concentricity to within 0·001 in total indicator reading.

"Eltropilot" plugboard for programming Müller multi-tool machines



Tapping rear driving member on Huller pitch-control machine



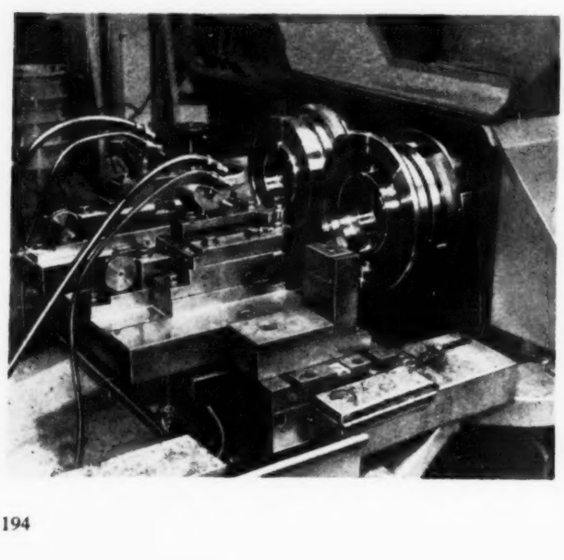
Typical operations on New Britain fine boring and turning machines. Left: front driven member. Centre: rear driving member. Right: rear bearing hub

A tooling layout drawing shows the fine-turning, facing, and grooving operations on the front driven member, with the spigot to receive the inner race of a ball-bearing held to a tolerance of 0·0005 in on diameter. Tool holders are pivot mounted and furnished with a micrometer adjustment. The lever ratio is 5:1, giving a tool movement of 0·0002 in for each graduation on the dial. A pair of driven members, front and rear, with different machining dimensions is handled at a single setting, and with spindle speeds of 1,180 rev/min the floor-to-floor time for a pair of components is 51·6 seconds. Air-mist cooling is provided to obviate the possibility of thermal distortion of the work, and all diameters are checked on Solex air gauges.

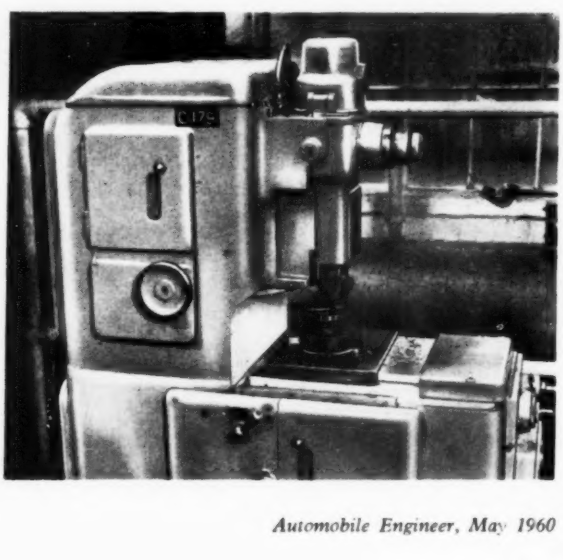
Other examples of work performed on these machines are illustrated. The rear driving member, chucked on its inner diameter and concentric within 0·003 in, is bored to a tolerance of 0·0009 in on diameter for the outer race of a ball-bearing, and faced parallel with the location face within 0·001 T.I.R. Spindle speeds are 1,181 rev/min and the floor-to-floor time for two components is 80·6 seconds. A smaller component, the flanged, registered, rear bearing member, is held concentric with the spigot diameter to within 0·001 in T.I.R. Bored and turned diameters are machined with tolerances of 0·001 in and faces in two planes are held square with the axis. Total time for two components is 41·5 seconds, giving a gross production rate of 173 units per hour.

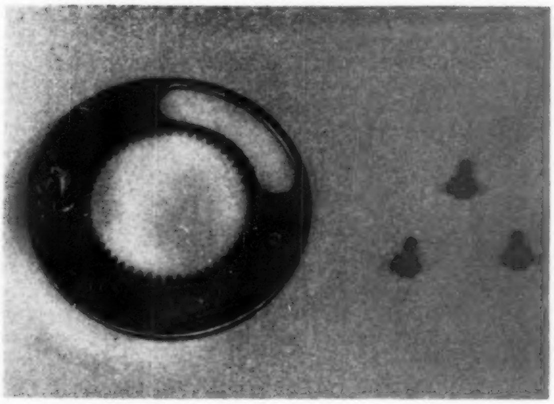
The main interfaces of the four rotating members are hard-chrome plated in a Canning semi-automatic plant located adjacent to the machining area. A continuous coating is applied to the specific surfaces; the thickness of the coating being 0·0005-0·001 in on the driven members and the rear driving member and 0·001-0·0015 in on the front driving member. Components are first degreased in a

New Britain machine fine boring front and rear driving members



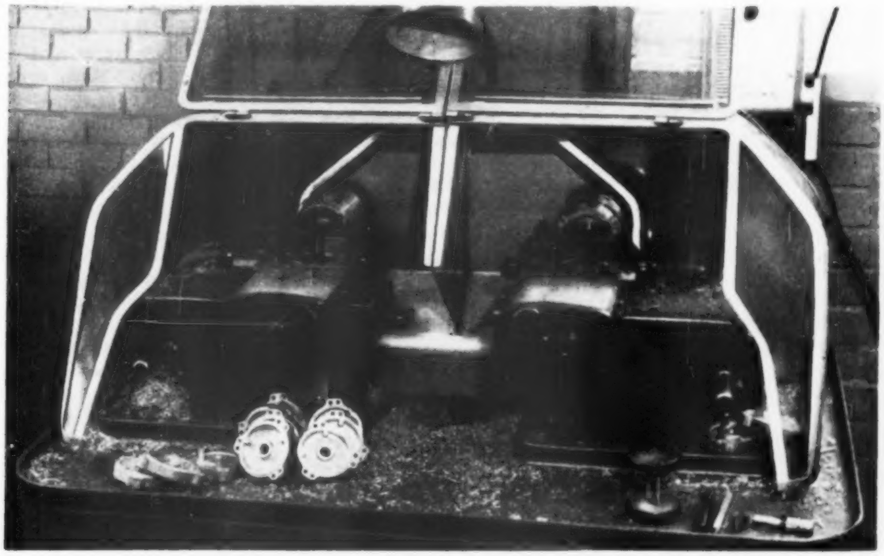
Governor annulus and pinions are cut on a Sykes hobber





Toothed annulus and planetary pinions for governor unit

Moulded assembly of governor permanent magnet and moving contact



Kummer twin-head, semi-automatic lathe for the high-speed machining of small components

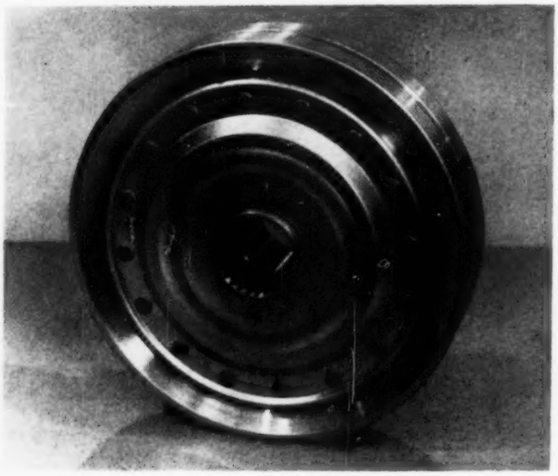
trichlorethylene bath and are then jigged in pairs. A coating of glycerine and French chalk is applied to the surfaces to be plated before the components are waxed in a two-stage process by being given a hot dip and cooling and repeating the operation. The treated surfaces are then stripped of the wax, scoured with pumice, wire-wool, and cloth, and etched in a chromic acid bath for one or two minutes. Parts are then transferred immediately to the plating bath for periods of 30-35 min or 45-50 min according to the thickness of coating required.

There are five plating units in the plant, each capable of handling ten of the main rotating components. Each unit can be individually controlled and current is held to 126 amp. Trivalent build-up is countered by periodic changeover of the use of individual plating units from driving members (internal plating) to driven members (external plating). After plating, the components are given two successive dip-tank rinses and a final cold-water spray rinse. Incidentally, the drag-out solution in the rinse tanks is utilized for topping-up the plating tanks. Components are then immersed in boiling water to remove the wax, which is floated off and collected for re-use. When cooled, the parts are de-jigged and finally cleansed by a vapour-fluid trichlorethylene treatment and drained. Thickness of coating is non-destructively checked by a Solus-Schall electronic instrument.

Smaller components requiring hard chrome plating are, of course, batched in numbers greater than ten. The governor plate, for example is handled with 24 mounted on a single jig.

All electrical components are made in the Witney factory, except for the relay which, however, will in future be also made there. Numerous small mechanical items are produced in a light machining section. The governor annulus, which carries the high-speed contact, may be cited as an example. Pierced and blanked from phosphor bronze strip, it is planished and then, on a Newall Rigidlap machine, is lapped to thickness. The centre hole is bored, and then the internal gear teeth are cut on a Sykes hobbing machine which is also used for the three small complementary planetary pinions.

Many instances occur of small components fabricated from pressed parts and then machined to final dimensions. This practice reflects the abundant press capacity of the company and its wide experience in the production of presswork. A cup-shaped solenoid housing is a deep drawing in steel, with a pressed steel flange projection-welded in position. On a Herbert No. 5 Preoptive turret lathe, this fabrication



Driven member assembled with Borg and Beck spring centre

is faced internally on the base and the open end is faced and screwed. The solenoid coil tube is pressed into an end cheek and rolled into a groove to secure by a capstan lathe operation. On the solenoid end plate a boss to receive the coil tube is knurled. After the knurling operation, 0·010 in is skimmed off to remove the high points of the knurled surface. This provides a greater contact area and a more secure lock when the tube and plate are press-assembled.

On the governor annulus a contact pillar with a knurled stem is given an insulated mounting with a nylon bushing in the serrated hole. The press-assembly of these parts is jigged to ensure the precise height and radial position of the contact. Other components are of composite construction with metal parts moulded in plastics material. Such items are produced on Bradley and Turton moulding presses, tooled for accurate location of the inserted metal parts. An example is the sintered, ring-type, permanent magnet assembly for the governor unit; the core and flange are of plastics material and the moulded-in contact pillar is formed of an extruded metal section.

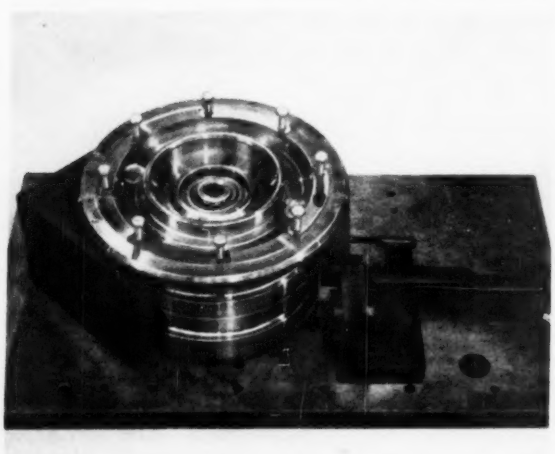
For machining the die cast aluminium end cover of the governor unit a Swiss-built Kummer semi-automatic, twin-spindle, front-turning lathe is used. This machine has individual, two-speed, motor-drive and control to each spindle and positive, cam-actuated, spring-returned tool carriages. Chucks are air operated, with pedal control valves. Spindles are tooled-up to machine different sides of the component and a semi-skilled operator merely unloads and reloads at one station while the other is machining and then repeats at the other station when the cycle is completed. Running at the lower spindle speeds of 900 rev/min, a cover is machined on both sides in 32 sec.

Other items precisely finished on the Kummer lathe are the carrier plate for the governor planetary pinions and the solenoid end cheek. The governor magnet assembly mentioned earlier is recessed on the plastics flange to provide a reference face for the drilling of the contact pillar. Incidentally, on these small components, all tapped holes in non-ferrous materials are drilled appropriately oversize to normal and tapped with Harris fluteless taps. These tools have been proved to have a longer effective working life, and to be less susceptible to jamming and breakage, than conventional fluted taps.

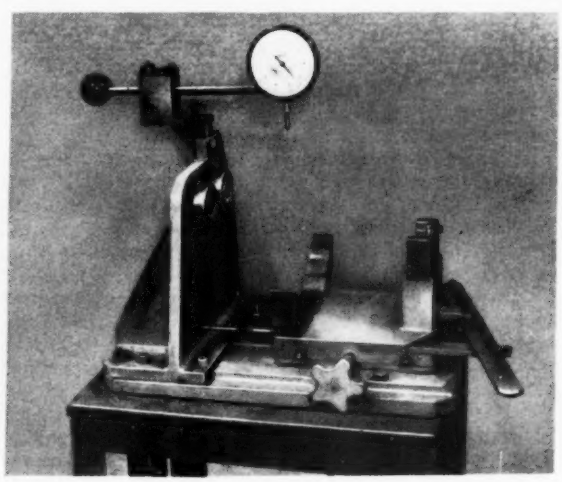
Assembly of the coupling unit commences with the fitting of Borg and Beck spring centre plates to the driven members. They are attached by 16 rivets that also secure the cover plate. The mating faces are coated with Goodyear Pliobond to keep out the magnetic clutch powder. Each member is then statically balanced on a Jackson and Bradwell machine, the maximum unbalance tolerated being 0·25 oz-in. On the other side of the member two labyrinth rings are fitted over a shoulder and secured by spinning in a Turner machine. They are again tested for balance, this time with a tolerance of 0·3 oz-in. unbalance.

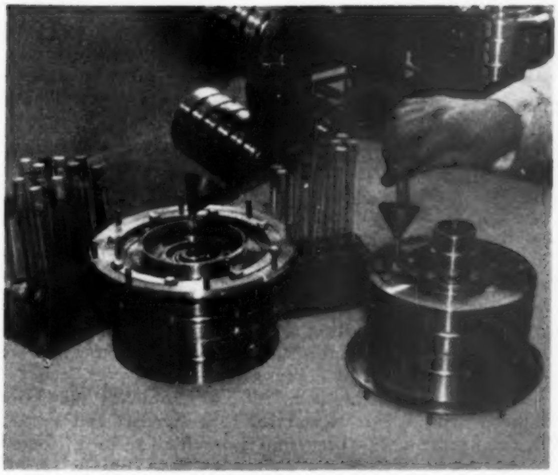
Driving members are then each fitted with two labyrinth rings, secured by spinning as before. Both driving and driven members are degreased by a vapour and liquid trichlorethylene treatment and subsequently a percentage check is made on these components on Hanovia fluorescent crack-detection equipment. In the rear bearing member a thrower ring is first inserted and over this the Hoffmann ball-bearing is pressed in, at a loading of 1,000-1,200 lb, on a Hi-Ton hydraulic press. An automatically operated valve on this machine lowers the working pressure to 500-600 lb for the following operation in which the rear driven member hub is pressed into the ball-bearing. The depth of the assembly is checked, a circlip is fitted in the grooved end of the hub protruding from the ball-bearing, and the

Clamping fixture for the concentric assembly of the rotating members



Inspection rig for checking the concentricity of the rotating assembly





Loading a measured charge of ferro-magnetic powder into coupling rotor assembly

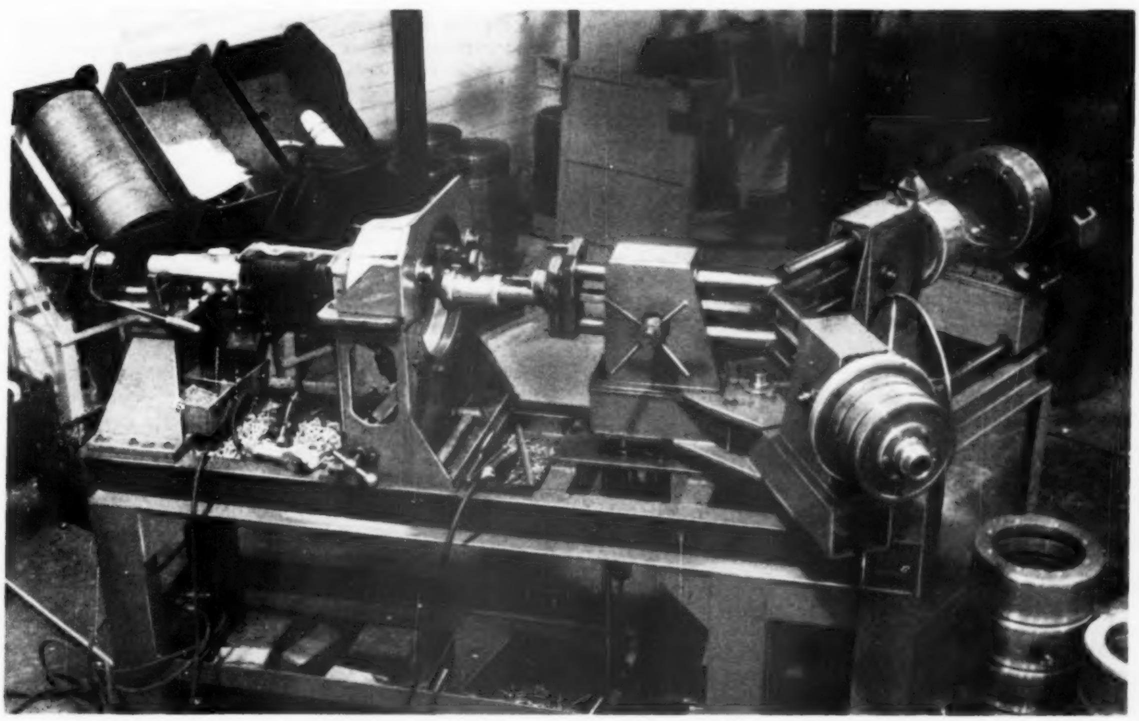
Dynamic balancing of charged rotating assembly on Jackson and Bradwell machine

assembly is racked. Assembly of the front driving and driven members is similar, with the exception that the ball-bearing is retained in the driving member by a circlip.

The two driving members are registered one on each side of a division plate and bolted up to form a single rotating unit which is subsequently registered and bolted to the engine flywheel. Of symmetrical section, the division plate is provided with two labyrinth rings on each face, secured by spinning over the supporting shoulder as on the other members. To ensure concentricity, the assembly is carried out on a fixture that aligns the parts on their outside diameters. The rear sub-assembly is placed on the plane surface of the fixture and against a large, vertically arranged vee block. Division plate and front sub-assembly are added and eight long set bolts are inserted. A presser block actuated by a toggle lever holds the parts into the vee block, clamping them securely while the set bolts are torqued in pairs to a predetermined value by a Desoutter twin-spindle nut runner.

The unit is then inverted and the rear hub centre, complete with oil seal ring, is registered in the driving member to secure the ball-bearing, and is bolted up with six set bolts. A check of concentricity is then made on the unit. It is mounted for rotation on a fixture and readings are

Indexing assembly rig for fitting stator and rotor elements of the coupling into the Rootes gearbox and bell housing



taken at four points on the outside diameter. Permissible tolerance is 0·003 T.I.R. Having passed this check, all the set bolts are locked up with twin tab washers.

Next, the complete rotating assembly is dynamically balanced on a Jackson and Bradwell machine. When mounted, the inner, driven members are held stationary by a flexibly connected mandrel engaging the splined hubs of the two members. A balanced driving fitment is secured to the front end and at the rear the unit is supported on the bearing diameter of the driving member hub centre. The machine gives a direct indication of out-of-balance values at the front and rear ends of the assembly and also the angles at which they occur. If necessary, any correction is effected by drilling in the flanges of the appropriate driving members. To facilitate re-assembly the three components constituting the driving unit are marked for radial alignment.

Extreme care is taken in the preparation of the powder charges for each coupling and the filling of the couplings. The ferro-magnetic powder is a chrome-iron alloy, sintered and processed to ensure consistency in particle size and shape. Both couplings receive an identical charge of 67 grams of powder from which all moisture has been excluded. Charges are weighed out, to an extremely close tolerance, in a temperature and humidity controlled room and stored in tightly corked test tubes. The powder is carefully introduced into the driving member of the coupling through a hole which is then closed by an aluminium plug. Both the plug and the attachment studs were, of course, fitted before the balancing was carried out. Finally, the completed and charged coupling unit is given an anti-seizing, anti-corrosion spray of molybdenum-disulphide over the entire external surfaces.

Three members, the central body and two end cheeks, are registered together and secured by socket screws to form the stator, which is registered in the coupling housing and held to the rear face by set bolts. The field coils are housed in the annular cavities in the assembly, each coil being packed with three undulating springs. Secured by screws to the front end cheek is a baffle ring which overlies the edge of the flange of the front driving member. Its purpose is to prevent burrs or metallic debris that might be detached from the starter ring gear gaining ingress to the interfaces of the stator and the driving assembly. The stator assembly is sprayed on its inner diameter with molybdenum-disulphide.

Completed stator and rotor units are loaded alternately on to an overhead drying and storage conveyor and transported to the rig on which they are assembled with the Rootes gear unit. The gearbox with its bell housing and rear extension, is received with a blanking cover closing the bell and a slave cover over the solenoid armature and aperture. After the covers have been taken off, the unit is mounted on the fixture, being located on the crankcase dowels, and supported at the rear. An air hoist is used to lift the stator from the conveyor and transfer it to the receiving table on the fixture. There, a capstan-operated carrier picks up on the stator bore. After indexing the fixture through 120 deg to align with the bell housing, the stator is fed into the bell by means of the capstan. Leads from the stator coils are threaded through an aperture in the bell. When in position, the stator is drawn up to a shoulder on the rear of the housing and fixed by set bolts. These bolts also hold a clip for the electrical leads which run under the box to the solenoid.

The fixture is again indexed and the unit advanced to locate on the dowelled centre. Here, a check of concentricity is made. Readings are taken on the three elements of the stator assembly, the tolerance of eccentricity being 0·005 T.I.R. After a further indexing movement, the rotor assembly is fed into the stator. Various items of control equipment are fitted and adjusted on the gearbox. Leads from the stator coils are fitted, two are earthed on the stator and two to different-sized terminals which receive tags from the bottom harness. These connections are fitted with the waterproof cover. The solenoid leads are taken to contacts which are enclosed by a metal cover. A rubber boot fitted over the cover is sealed by a Jubilee clip.

Every gear and coupling assembly is given a functional test on a specially developed test rig. A slave flywheel, complete with jaw clutch to take the drive, is fitted to the coupling. The complete assembly is hoisted and mounted on the test bed, located on the crankcase dowels. The harness is plugged in from the electric control unit of the assembly to the control console. The drain plug is removed from the gearbox and a connection is made through which an intermittent, pulsating, oil supply is delivered into the box. This method saves time that would otherwise be occupied in filling and draining the box.

A 75 h.p. constant-speed A.C. motor drives the transmission unit, which is coupled to a Heenan and Froude dynamometer, at 3,000 rev/min throughout all tests. Control of the unit on test is from a remotely sited console. A run of one minute duration is made to distribute the powder in the couplings. Then, first, second and third gears are selected followed by a run through third, second and first gears. Next, third speed is selected with a limited torque on the front clutch, followed by the selection of second gear with a limited torque on the rear clutch. During these latter tests each cluch is checked for torque capacity, which must lie within certain predetermined limits. Finally, a check is made in the de-energized condition to ensure that the drag on each clutch is at an acceptably low level. Under normal conditions, the approximate time for a full check is from 6 to 8 minutes.

On leaving the test bed, any necessary adjustments are made to the front coupling. Should this component have too high a torque capacity, a carbon resistor is inserted in the feed to earth. If the torque capacity is too low, the test has failed and the unit is taken back for rectification. The slave flywheel is removed and the blanking plate is replaced. A guard to protect the solenoid from debris thrown up from the road is fitted and various apertures are closed. For road transport back to the car manufacturers, three transmissions are carried in each wooden pallet.

Strathclyde PA-10 Primer for Aluminium

BECAUSE of its smooth surface, aluminium is a relatively difficult metal to paint. Its natural lack of "key" has always presented a problem of adhesion. In the past, various methods of improving adhesion of the primer coat have been used. The surface of aluminium has been etched by one of the pre-treatment processes or roughened by the vigorous use of an abrasive. Chemically active primers replaced these expensive methods but, because of the chemical effect of etching acid on varnish mediums, most of the primers offered were of the two-can type.

One of the first etching primers to be produced ready for use in a single can was the outcome of extensive research conducted by Federated Paints Ltd., of Dobbies Loan, Glasgow, C.4, and the latest version of this product is known as Strathclyde Etching Primer PA-10. For this, no special preparation of the metal surface, other than normal cleaning, is necessary. A single coat of PA-10 will give an adhesion of over 3,000 lb/in^2. In addition to possessing this remarkable adhesion, the primer is fully pigmented and provides an effective barrier against corrosion and weathering of the aluminium surface. Even after long periods the primer still provides a solid foundation for the reception of subsequent coats of any normal paint system.

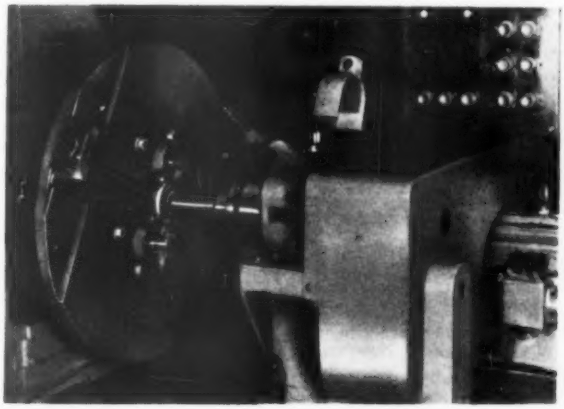
GRINDING BIG-END BORES

Newall-Keighley Automatic-cycle Internal Grinder for High-rate Production of Connecting Rods

BASICALLY, this heavy-duty, fully automatic machine, incorporating both gauging and diamond sizing mechanisms, is designed for grinding holes of from 0·25 in to 6·0 in diameter. On the model AK illustrated, specifically intended for grinding connecting-rod big-end bores to receive the bearing bushes, the drawbar chuck supplied for general-purpose duties is replaced by a special workholding fixture designed to ensure the absolute parallelism of small-end and big-end bores.

Mounted for endwise location on the T-slotted table, the wheelhead is generously proportioned to accommodate interchangeable wheel spindles to cover a wide range of speed requirements. Spindle drive from a 3 h.p. motor running at 3,000 rev/min is by an endless cotton belt, adjusted for tension by a jockey pulley. The table slides on automatically lubricated vee and flat guideways and, in addition to a rapid traverse rate of 420 in/min, independent and adjustable speeds are provided for roughing, finishing, and wheel dressing. Together with traverse speed, these are controlled by grinding and dressing dogs and an adjustable cam mounted on the front of the table. Minimum and maximum table strokes are 0·031 in and 6·0 in respectively.

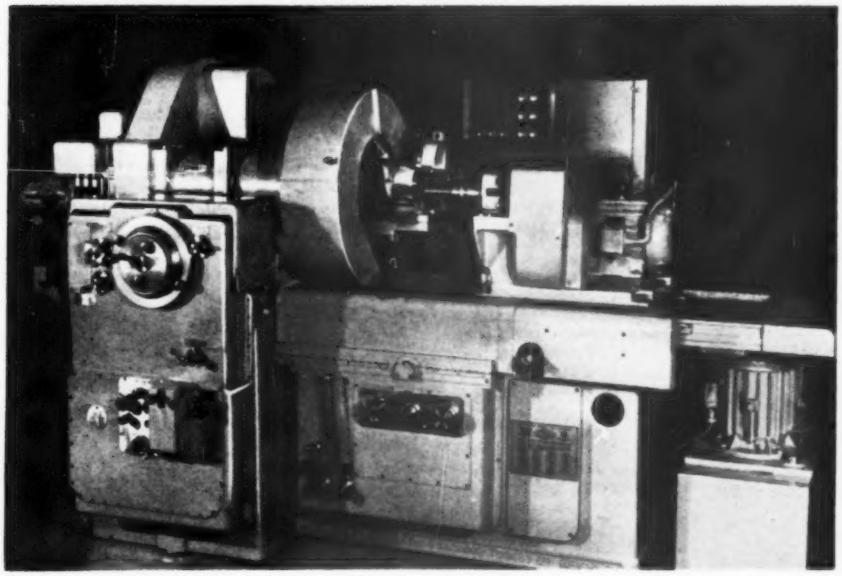
The workhead carrying the workholding fixture is mounted on the feed cross-slide and also accommodates the gauge sizing mechanism. A 3·0 in bore flanged spindle runs in Gamet high-precision, double-row, taper-roller bearings, and work speeds up to 1,000 rev/min are available. To minimize slowing-down time, a D.C. injection brake is incorporated, and built into the spindle is a hydraulic fixture-operating mechanism with a 2·0 in maximum stroke and adjustable pull or push pressure. The maximum power cross feed is 0·042 in on diameter, and one revolution of the handwheel control gives an infeed of 0·10 in on diameter.



Entirely independent of table traverse, the continuous cross feed is operated by a cam and roller mechanism, and is positioned by a screw and nut. This screw also serves to compensate automatically for wheel truing. The hydraulically operated feed cam controls both rough and finish grinding rates, and is automatically locked during the dressing feed stroke. A range of cycles to meet requirements is obtainable by adjustment of the feed controls, and automatic feed may be applied or disengaged during the grinding cycle.

Also hydraulically operated are the double-gauge sizing system and the wheel-dressing attachment. To correlate travel of the diamond with movement of the table, advance

(Above) View of the special workholding fixture, wheelhead, and wheel dresser



Newall-Keighley AK internal grinding machine arranged for the rapid production of connecting rod big-end bores

of the dresser arm is controlled by a servo-mechanism actuated by a cam on the machine table. A micrometer adjustment is provided for the dresser.

After initial setting-up, the machine is controlled by manipulation of a single lever. The worktable advances at rapid traverse speed until the grinding wheel enters the bore, then the rate is reduced to grinding speed and the table reciprocates on a stroke adequate to traverse the entire depth of the bore. At this stage, roughing feed begins and is continued until a predetermined diameter is reached and the appropriate gauging member is able to enter the bore.

Continuing the automatic cycle, the wheel backs off, then retracts from the bore, and the table speed is reduced to that for dressing as the grinding wheel approaches the diamond position. To ensure that the wheel is fully trued, dressing takes place during both outward and inward strokes. On completion of dressing, the diamond withdraws, the wheel re-enters the bore and regains contact. Table reciprocation continues at a lower speed, fine feed is engaged, and the finish gauge enters the bore as finish size is attained. Then the gauge withdraws, the wheel backs off and leaves the bore, the feed dial retracts, and the table returns to the loading position. Compensation for wheel wear is effected automatically at the termination of each grinding cycle.

An indication of the output potential of the machine is given by the following production times maintained under normal working conditions. They relate to components bored concentrically and requiring 0·014 in stock removal:

Bore finish diameter	2·394—2·395 in
Bore depth	1·515 in
Rough grinding	15 sec
Wheel dressing	7 sec
Finish grinding	7 sec
Loading and unloading	25 sec

Newall-Keighley grinders are manufactured by Keighley Grinders (Machine Tools) Ltd., and are marketed by Newall Group Sales Ltd., of Old Fletton, Peterborough.

Protective Packing

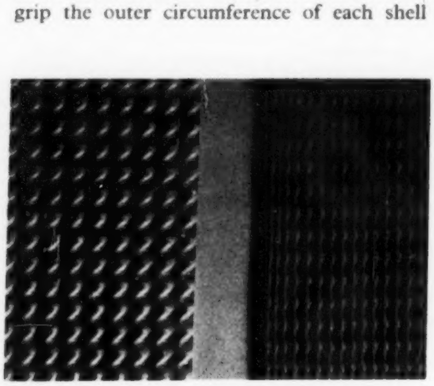
Special Material and Methods Used for Vandervell Thin-wall Bearings

THIN-WALL bearings present a difficult packaging problem. On the popular copper-lead-indium bearing, for example, the vital layer of indium may be less than 0·001 in thick. Special care must be taken to ensure that no superficial damage occurs during transit or storage. The method adopted by Vandervell Products Ltd., for their larger bearings, is of interest and is applicable to the packaging of other precision components. The material used is Carbion, made by Spicers Ltd., at Loughton, Essex, by passing strawboard through heated rolls. A permanent three-dimensional pattern is imparted, as shown on the left of the illustration, and the resulting material possesses a certain resilience. If the delivery of strawboard from the machine is restricted, the pattern is telescoped into closer ridges, as shown at the right. In this form the material can support considerable loads without flattening, and it can also be stretched. This feature is exploited by forming the flexible material into tubes or sleeves and by pulling them over the components to be protected.

Vandervell use flexible Carbion sleeves heavily impregnated with a beeswax compound. Bearing half-shells are first dipped in molten beeswax, a basket of bearings being allowed to soak at about 200 deg F, so that the wax drains off cleanly, leaving a smooth, thin, adherent layer. The shells are then individually inserted into the sleeves, which grip the outer circumference of each shell and span the diameter. Shells are nested in cardboard cartons so that little space is wasted. In this way, any shocks suffered during transit are absorbed by stretching of the Carbion sleeve between the bearings, and the internal surfaces never contact other bearings or the packing material.

This packaging system was first instituted for shipment of bearings to the manufacturers of Cummins diesel engines in the U.S.A. Measuring up to 4½ in diameter, these bearings were difficult to protect because their weight caused high inertia loadings when subjected to shocks in transport. Since the Carbion system was introduced, not one case of damage has arisen, though more than one million bearings have been exported. The double beeswax protection eliminates the risk of corrosion during the long sea voyage, and is stated to ensure a safe shell life of about five years.

This simple and relatively inexpensive packing material has many possiblities in the motor industry. One of the most likely fields of application is in the making up of CKD shipments, where machined components and complicated pressings are nested in crates in considerable numbers. The standard resilient Carbion readily conforms to double-curvature shapes such as lamp reflectors, bumpers and body panels. Flexible Carbion sleeves are particularly suitable for the smaller, precision-machined or highly finished components. Apart from the protection afforded, the new system effects economies in space, labour, and materials.



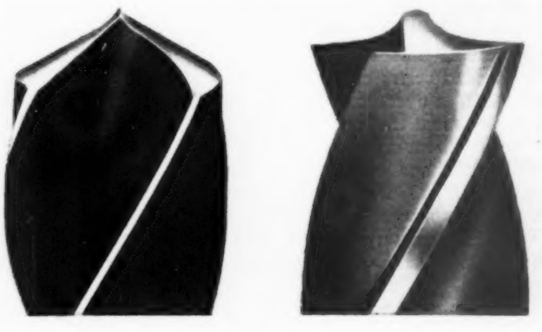
Standard Carbion (left) has excellent cushioning and shock-absorbing properties. Flexible Carbion (right) has good crush-resistance and is stretchable

Sleeves of flexible Carbion are stretched over the bearing halves, bridging the diameter. The shells are nested so that shocks are absorbed by stretching of the Carbion

THE SPIRAL-POINT DRILL

A New Form of Drill-point Geometry Developed by the Cincinnati Lathe and Tool Co.

APART from minor modifications there has been virtually no change in the form of the conventional twist drill for many decades. The spiral point, now fully developed by the Cincinnati Lathe and Tool Co., of Cincinnati, Ohio, U.S.A., represents a major advance. Research showed that several disadvantages of the conventional drill stemmed from the almost flat chisel point and that this point was the feature most needing investigation and also most likely to yield worthwhile improvement in performance. It required a new conception of drill-point geometry to eliminate the chisel point and provide a relatively sharp spiral point.



A spiral-point drill in its standard 118 deg form (left) and the special 180 deg form for sheet metal work (right)

Further, and scarcely less important, it necessitated the design and development of a machine to grind the new point accurately, symmetrically, and economically. Obviously, it could not be ground free-hand or on an orthodox drill grinder.

Significantly, only the point is affected; the new point is formed on an otherwise standard, conventional twist drill. Thus use of the spiral-point drill does not necessitate change of equipment or of tooling set-up, although it offers the possibility of simplifying tooling or even eliminating certain operations. In comparison with a conventional chisel-point drill the spiral-point drill, it is claimed, has many advantages:

(1) Can reduce or eliminate centre punching, centre drilling, and guide bushing
(2) Has no tendency to wander, either on the surface of work or in the hole
(3) Produces a more truly circular hole that is closer to size
(4) Thrust and torque required are reduced, giving increased drill life and fewer breakages
(5) Cooler operation owing to less thrust, and consequently less risk of work distortion
(6) Reduces the need for secondary operations such as chamfering or reaming
(7) Produces in sheet metal a round hole free of burrs. Has no tendency to bite into the work on the break-through.

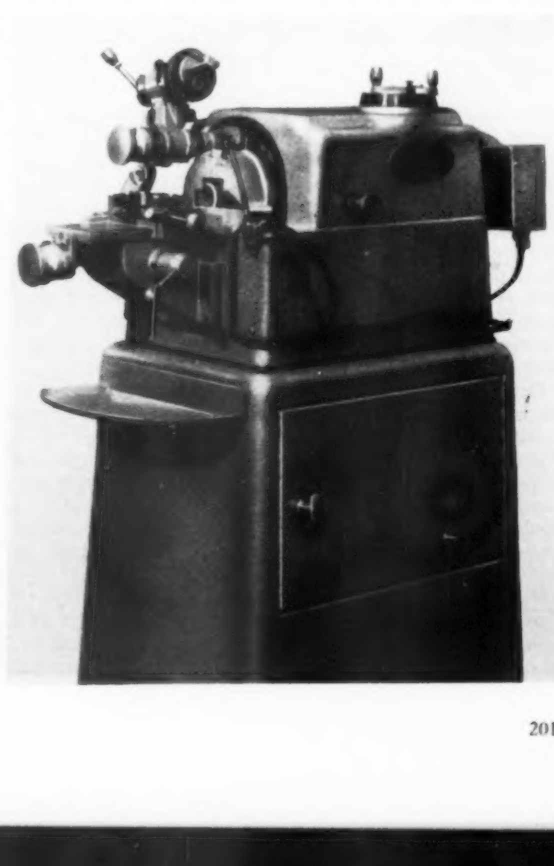
As regards hole location, the ordinary chisel point presents a straight edge normal to the drill axis, and parallel to a flat work surface. Unless the drill point is ground precisely symmetrical it tends on rotation to describe a planetary motion and wander from the intended location. This tendency must be restrained by centre punching, drilling, or adequate guide bushing. With the spiral-point drill a sharp point is presented to the work and it automatically centres itself.

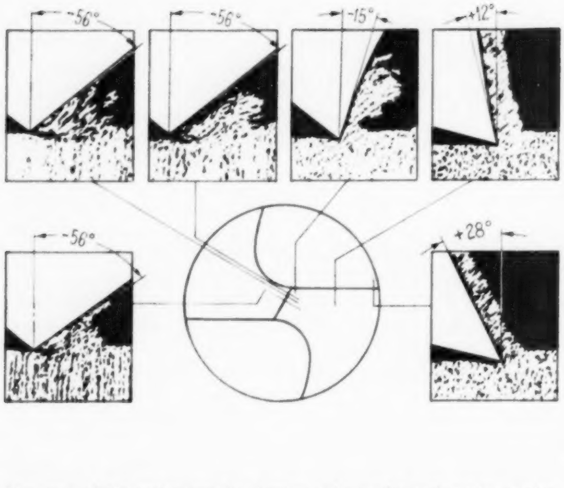
This tendency of the chisel point to wander has further disadvantages. As is commonly revealed in the depression formed before the drill has cut to full diameter, the shape of the hole is polygonal rather than circular. As drilling is continued, the flanks of the drill tend to correct the shape but in many instances cannot achieve a complete rectification and produce a truly circular hole. This is the more marked in shallow holes or thin workpieces.

Apart from the shape of the hole, size is also affected adversely. The slightly mis-shapen hole is drilled oversize and the loss of dimensional accuracy persists to some depth. Inspection will disclose the familiar "bell-mouthing" at the entry face of the work. A spiral-point drill, by contrast, cuts a circular hole of predictable and dependable diameter. Bell-mouthing is virtually eliminated.

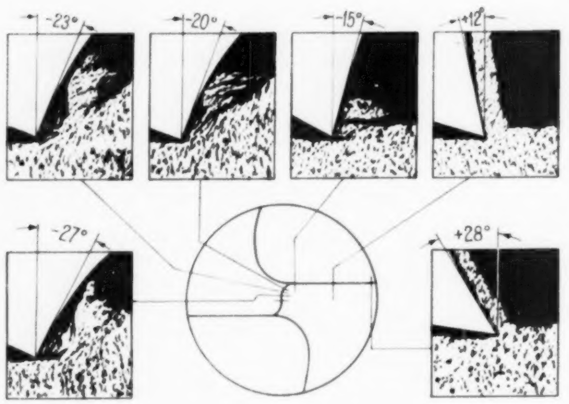
It is this characteristic of non-circular, bell-mouthed, and inconsistently oversize holes produced by the chisel-point drill that necessitates a secondary truing and sizing operation of reaming or boring. In many instances, use of a spiral-point drill will provide consistently a degree of accuracy that will obviate the need for a sizing operation and consequently reduce both cost and production time.

The Cincinnati "Spiropoint" drill-sharpening machine. The grinding wheel spindle is parallel to and gyrates around the axis of the stationary drill





Diagrams of photomicrographs showing rake angles and cutting action at successive planes along the cutting edge of a chisel-point drill (above) and a spiral-point drill (below)



For deep-hole drilling, the spiral-point drill performs to marked advantage over the chisel-pointed tool. The self-centering action of the spiral point keeps the drill on location and produces a straight hole. In the drilling of hollow shafts from both ends the superior performance of the spiral-point drill is most marked. An incidental feature of some importance in deep-hole drilling is that the form of the spiral point gives an easier chip flow into the drill flutes. This reduces the number of backing-off strokes to clear chip congestion, with a consequent saving of cycle time. It also lessens the risk of drill breakages; not infrequently a serious item in respect of cost and production.

Drill life is greatly affected by the heat generated by friction in operation. Considerable effort is made to deliver coolant fluid, mist or air, to the point of operation and, in general, the cooler the drill the longer its life will be. Should the drill overheat, the cutting edges dull more rapidly and more frequent regrinding is necessary. This involves a double penalty as it increases down time for drill changes and increases the consumption of drills. Unless drills are maintained in good condition the accuracy of the work will suffer and the risk of drill breakage, and further loss of time, will be heightened.

A special study of the generation of heat when drilling was made by the Cincinnati Co. In the case of the conventional chisel point the major problem was found to be the lack of cutting action at the point, which has a high negative rake angle. Although in some metal-cutting operations, conducted under certain conditions and at high speed, good results are obtained with negative rake, at the point of a drill conditions are unsuitable and the speed is very low. Hence, it was concluded, a radical change in point geometry was required.

Pursuing these investigations a series of photomicrographs were obtained of 0·500 in diameter drills in operation. The method employed was to bring the drills to an immediate dead-stop whilst they were forming chips. Both drills and workpieces were then sectioned in planes at different radial distances from the drill axis. These sections were photographed and enlarged.

The diagrams, prepared from the photomicrographs and reading clockwise from the left, relate to section planes at 0·010, 0·020, 0·030, 0·050, 0·140 and 0·240 in distance from the axis. From zero to 0·030 in from the axis the rake angle is minus 56 deg. Thereafter, the angle improves and eventually changes sign. Comparison of the chip structures in the first and the last of the diagrams provides evidence of the inefficient cutting action of the point. There the metal is not being cleanly sheared off, but rather is being pushed or rubbed off with heavy friction and the generation of considerable heat.

A second series of diagrams shows sections of a spiral-point drill in operation; the sections being taken at the same distances from the axis as those for the chisel-point drill to permit direct comparison. The three relevant sections at 0·010, 0·020, and 0·030 in from the axis show negative rake angles of 27, 23, and 20 deg respectively. The figures are a manifest improvement and the diagrams indicate that positive cutting action and chip formation commences at the drill point. In such circumstances, less frictional heat is generated.

As a consequence of improved cutting action, the thrust required to force the drill into the work is lessened; by from 15 to 34 per cent, with the greater reduction being obtained at the slower feed rates. Driving torque is also reduced, in some instances by as much as 15 per cent.

For sheet-metal work, as in body production, the spiral-point drill is of particular value. Its self-centering action eliminates the need for centre spotting and the hazard of producing surface-damaging runs through inadvertent mis-starts of drilling. Drills intended for sheet-metal work are specially ground to an angle of 180 deg instead of the normal 118 deg, as shown in the illustration. A familiar problem when drilling sheet metal, particularly when drilling with hand tools, is the tendency of the conventional drill to bite into the metal as it is breaking through. This usually results in a hole that is badly out of round and with large and ragged burrs on the underside of the work. It is also a frequent cause of breakage of small-diameter drills. The spiral-point drill ground to 180 deg obviates this difficulty. It cuts a circular hole and the outer lips act in the manner of a trepanning tool, leaving the hole practically free of burrs on both sides of the work.

Grinding of the spiral-point drill must be precise and the special "Spiropoint" machine is an essential. On this grinder the drill is held stationary and the grinding wheel makes a gyratory motion around the axis of the drill. Operation is simple, with dial setting for drill size and angle. Feed is by hand in increments of 0·001 in for dry grinding. A mist coolant system is available if faster stock removal and better finish is required. For the smaller plants, for which a machine would be uneconomical, drills can be resharpened by tool factors as a service.

Charles Churchill and Co. Ltd., Coventry Road, Birmingham, 25, are selling agents in the United Kingdom for the "Spiropoint" grinder and other Cincinnati tools.

New Plant and Tools

Recent Developments in Production Equipment

In the range of Johansson cylindrical grinders, handled in the United Kingdom by Stuart Davis Ltd, Much Park Street, Coventry, are two universal machines 1U-A and 2U-A having swings of 10 in and 14 in respectively. Model 2U-A, illustrated, is available in four versions of different length to handle work up to 78 in between centres. The wheelhead will swivel through 360 deg and can be used at either the right or left of the table for plunge, shoulder, or diameter grinding. Twelve speeds ranging from 20 to 330 rev/min are provided for the workhead. A live spindle is standard equipment but a dead centre is supplied on demand. Chucks, collets, and centres are all interchangeable at both ends of the spindle which runs in super-precision roller bearings.

As standard, the tailstock centre is manually retracted, but it can be hydraulically operated and foot controlled if desired, to leave the operator's hands free for loading and unloading. Since the tailstock is not reversible, a special tailstock is required for operations at the end opposite to normal. A hinged internal-grinding attachment, shown fitted, is suitable for grinding bores from ½ in diameter upwards. Should a specially high degree of surface finish be required, the normal wheelhead can be removed and replaced by a heavy, rigid, internal-grinding head.

The standard wheel-dressing attachment is fitted to the tailstock, enabling the wheel to be dressed without removing the workpiece. Other attachments are available; one for dressing both internal and external wheels and one for radius form dressing. Another fitment is a profile truing device, on which both the diamond and the tracing pin are set at an angle of 25 deg, so that it is unnecessary to change the position of the diamond, even when truing a radius on a 90 deg shoulder. This device is operated from the central hydraulic system. No pressure is exerted, however, on the diamond slide when in operation; only the weight of the vertical slide loads the diamond.

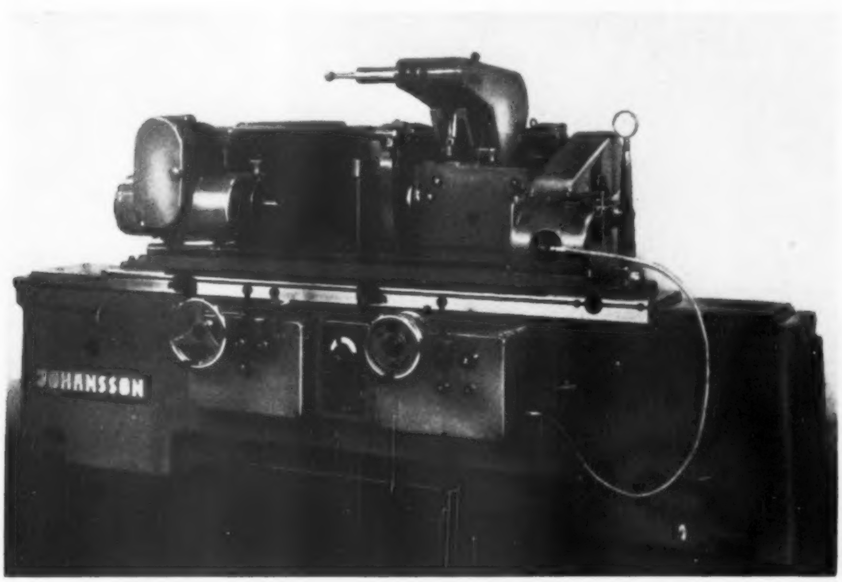
A 40 gal coolant tank is provided, divided into a number of settling chambers. The coolant pump is connected to the workhead motor so that both operate simultaneously. Electric motors are dust-protected and are furnished with individual fuses. All electrical controls are enclosed in a cabinet at the rear of the machine.

Vibration limit controller

With this instrument, operated from 110 or 200-250 V, 50-60 c/s mains and consuming only 40 W, vibration in machines or structures can be measured, monitored, and recorded. Manufactured by Dawe Instruments Ltd., 99 Uxbridge Road, London, W.5., the overall dimensions are 10 in × 14 in × 9 in high and the weight is approximately 24 lb. The pick-up, without probe, is 1¾ in diameter and 2¾ in high and weighs only 1 lb. Rotating machinery or heavily loaded structures are automatically monitored for excessive vibration by comparing the signal derived from the pick-up continuously against a pre-set bias voltage. A trip valve is triggered if the signal rises above the predetermined level and a heavy-duty relay then operates an alarm or shuts down the machine.

Of the moving-coil type, the pick-up is hermetically sealed to maintain accuracy under conditions of high temperature and humidity. Twin screened leads up to 200 ft long enable the controller to be placed at a safe distance from a pick-up mounted in a dangerous or inaccessible location. Over the frequency range from 20 to 400 c/s (1,200 to 24,000 rev/min) the pick-up provides an output voltage proportional to the velocity of vibration of the surface with which it is in contact. A signal representing displacement is obtained by applying the output voltage to an integrating network, followed by amplification and rectification. The resulting D.C. signal operates a meter calibrated in peak values of displacement for a simple harmonic motion applied to the pick-up. This meter reading of average vibration level can be recorded by plugging in a standard D.C. milliammeter, as in usual practice.

Johansson universal grinding machines
(Stuart Davis Ltd.)



Vibration limit controller
(Dawe Instruments Ltd.)

A sensitivity control provides four ranges in British or metric denominations: 0-0·001 in, 0-0·003 in, 0-0·010 in, and 0-0·030 in; or 0-0·03 mm, 0-0·1 mm, 0-0·3 mm and 0-1·0 mm. The controller can be set to trip at any vibration level between 20 per cent and 100 per cent of the full-scale deflection of the meter on any range.

Extrusion of sealant

An air-operated Graco pump mounted on a pneumatic ram is used to distribute and extrude Sealastic, used for windscreen, rear light, and weather assemblies, at Vauxhall Motors, Luton. The problem of dispensing this heavy material rapidly and conveniently at several stations on a body line has thus been neatly solved. This method enables a heavier-bodied material to be used than would otherwise be possible, which renders the operation quicker and cleaner.

The material is supplied in 40-gal, open-top drums and these are jacketted with Isopad heater units. Operating on an air pressure of 75 lb/in², the pneumatic ram is used to elevate the pump when loading and unloading drums, and apply a steady pressure to the ram in the drum when the pump is operating. The ram seals the material from dirt or foreign matter and scrapes the drum clean with negligible loss. Material is lifted directly from the drum, pumping being effected from the top to obviate aeration and channelling of the material.

Five pumps are used, each mounted on a pneumatic ram. The pumps have a pressure ratio of 40: 1, so that a line pressure of 75 lb/in² yields a material pressure of 3,000 lb/in². The material is pumped direct from the drum through ¾ in bore flexible hose to an overhead steel manifold, 1½ in bore × 30 ft long. From this manifold, take-off lines are dropped, each comprising 16 ft of ¾ in hose, plus a 6 ft tail piece of ½ in bore to give added flexibility, and terminating in an extrusion gun. Four guns are taken off each system, which is independently operated by one pump.

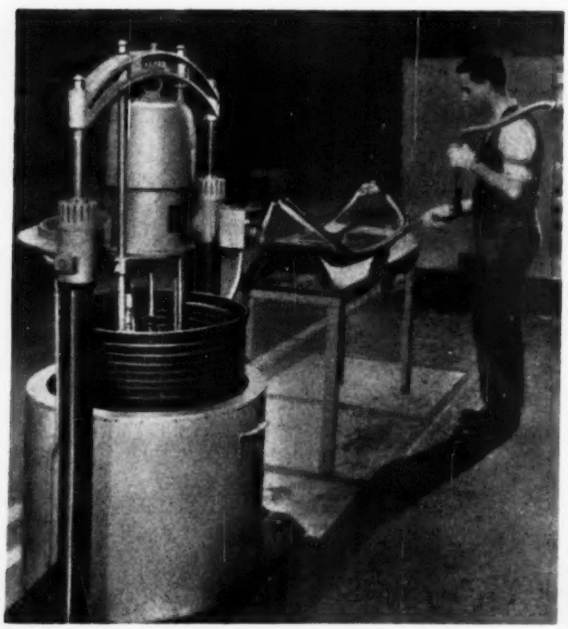
Sole distribution of Graco equipment for the United Kingdom is through Alfred Bullows and Sons Ltd., Long Street, Walsall.

Optical projector

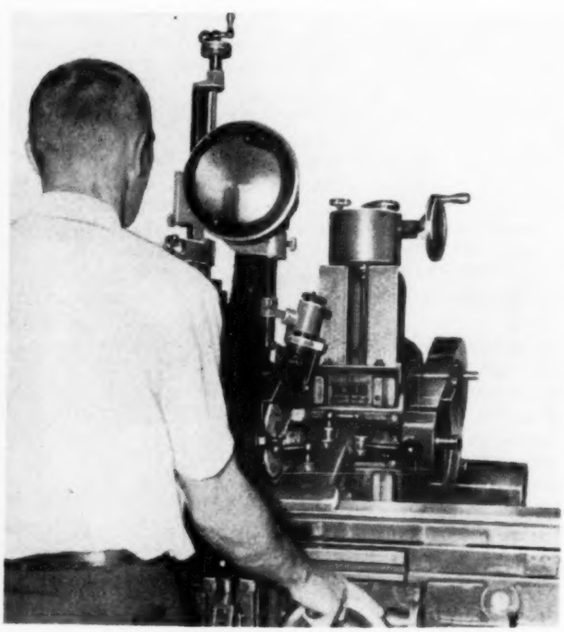
This portable optical projector may be attached to a variety of machine tools. It may be permanently mounted on existing grinders, lathes, milling machines or other tools, or can be set up when required in a period of minutes, using either universal or special adapters. In use it makes possible continuous, precise, magnified checking, direct measurement, and comparison of work contours against transparent overlay charts. It is readily converted from one application to another.

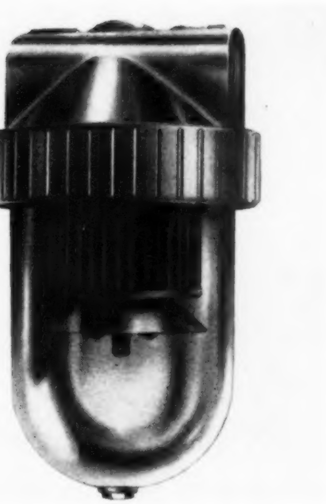
The instrument has found its most common application with circular and flat-form tool grinding, production grinding of precision contours, regrinding of tools, and dressing of critical wheel and tool shapes. It enables worthwhile savings in machining time, and consequently better use of equipment, to be effected together with closer control, less spoilage, and lessened operator training time. Simul-

Air-operated sealant pump
(Alfred Bullows and Sons Ltd.)



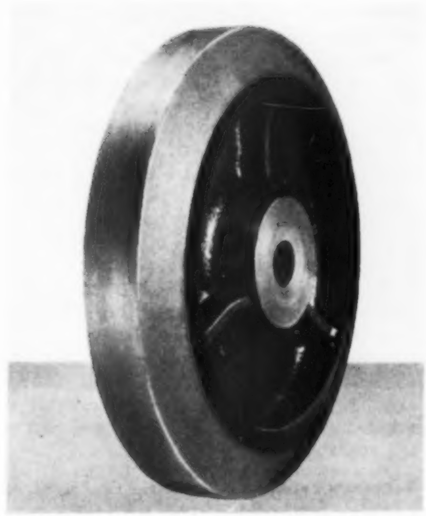
Portable optical projector
(Stocker and Yale, Inc.)





Left: Transparent-bowl filters for air lines
(Vokes Ltd.)

Right: Revvothane-tyred wheels
(Revvo Castor Co. Ltd.)



taneously, it provides a standing visual check on spindle and wheel alignment and runout conditions.

Standard interchangeable optical units give magnifications of 10X, 20X, 31·25X, 50X, 62·5X, 100X. Other available accessory equipment includes a protractor screen, illuminators, and compound mounting devices for direct linear measurements to 0·001 in or 0·0001 in. The manufacturers are Stocker and Yale, Inc., 40 Green Street, Marblehead, Massachusetts, U.S.A.

Compressed air line filters

From a study of air line filters conducted by Vokes Ltd, Henley Park, Guildford, Surrey, the conclusion was reached that for efficient operation, coupled with low resistance to flow for long periods between replacement, two types of element were necessary. Accordingly, a new range of transparent-bowl filters for $\frac{1}{4}$ in, $\frac{3}{8}$ in, and $\frac{1}{2}$ in air lines is designed to accommodate alternative filter packs.

A standard general-purpose element, VAF 57, is suitable for the removal of water, oil mist, or grit where the concentrations are average. The alternative element, "D" Pak, is for applications where oil or water mist removal is of primary importance and a high degree of purity is essential.

A single wing nut attaches the interchangeable element to the diecast light alloy housing and a ribbed clamp nut secures the transparent bowl, which is available in two different capacities. Standard units are capable of operation at pressures up to 150 lb/in^2, and for higher pressures up to 250 lb/in^2 metal bowls are supplied. To service it is merely necessary to remove the bowl plug and draw off the collected moisture and impurities. For optimum performance it is recommended that the element should be replaced at six-month intervals.

Work- and lift-truck tyres

Wheels for work trucks, and trolleys, and lift-trucks are now available with resilient tyres of polyurethane rubber. Designated "Revvothane" by the wheel manufacturers, Revvo Castor Co. Ltd, Old Ford Road, London, E.3, this material is claimed to have a working life ten times that of natural rubber. Load carrying capacity of tyres of Revvothane is approximately four times that of normal rubber tyres of the same size. The tensile strength of the material is of a high order, 5,000 lb/in^2, and this combined with good bonding characteristics gives a tyre that is not susceptible to incidental damage from debris encountered on the workshop floor, or rapidly become cut, torn or chipped.

To meet specific requirements the material can be cured to give any hardness from 65 deg to 95 deg Shore. Standard tyres have a hardness of 92 deg Shore, which precludes the "pick-up" and the "carry" of metal objects from place to place. Abrasion resistance is of the order of 30 to 40 mm^3, that is, three times that of natural rubber. Other advantageous characteristics, of particular value in workshop applications, are that it is unaffected by contact with oil or fats, and is highly resistant to high-aromatic motor fuel and aviation kerosine. Water absorption is nil. These new tyres can be used in conditions where natural rubber tyres would quickly become swollen and deteriorated.

They are but little affected by atmospheric and temperature conditions. They can be used in temperatures from —20 deg C to +120 deg C. They are quite unaffected by ozone or oxygen and therefore they never "age" or "perish" and their tough resilience is claimed to be retained indefinitely.

Dressers for coated abrasives

Dressing tools specifically developed for use on coated abrasive belts, discs, and pneumatic wheels are widely used in the U.S.A. The range of tools produced by the Desmond-Stephan Manufacturing Co. of Urbana, Ohio, is now being distributed in the United Kingdom by the Addison Tool Co. Ltd., 28 Marshalsea Road, London, S.E.1. Three

Dressing tools for abrasive belts
(Addison Tool Co. Ltd)

models are available, two with flexible-tooth cutters and the third with stiff-tooth cutters.

Selection of these tools is determined in accordance with the grain size of the coated abrasive to be treated. Model BCA-1 with flexible-tooth cutters $1\frac{3}{8}$ in diameter and a face width of $1\frac{1}{2}$ in is for use on abrasives of 200 to 300 grain size, while BCA-2 of similar dimensions is for abrasives of from 60 to 200 grain size. Both tools are suitable for either wet or dry applications. For abrasives of 60 grain size or coarser, the stiff-tooth dresser BCA-4 with cutter $1\frac{1}{4}$ in diameter and a face width of $\frac{13}{16}$ in is recommended. This, as a rule, is used on dry applications.

When dressing, a light infeed pressure and from two to four passes across the face of the belt should be sufficient. The dresser should always be applied with the cutters running parallel to the axis of the belt support rollers or, in the case of a disc, aligned on a plane intersecting the axis. A shearing action is undesirable and should be avoided. Cutters are easily changed and replacement sets are available.

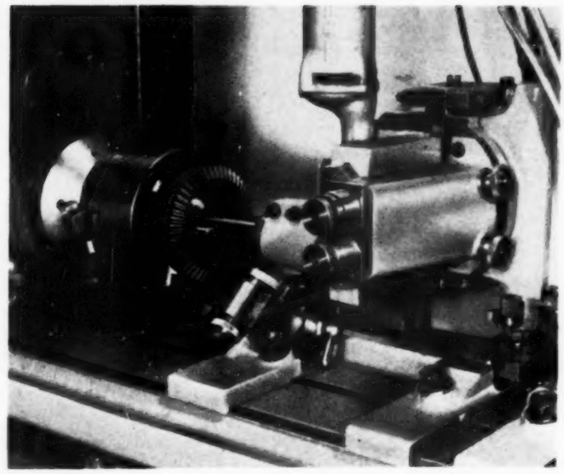
Vibratory finishing machine

A range of machines for finishing parts in the mass by a vibratory process has been developed in the U.S.A. by the Pangborn Corporation. This system makes possible the processing of interior or shielded surfaces that are inaccessible by conventional barrelling methods. American experience and technique is now available to the Hepburn Conveyor Co. Ltd., Rosa Works, Wakefield, which has recently concluded an agreement with the Pangborn Corporation to manufacture the machines and ancillary equipment in Britain.

An advantage claimed for the Pangborn machine over more orthodox tumbling or barrelling equipment is that both the amplitude and frequency of the vibratory motion can be varied. Thus each machine is capable of handling a wide range of work at optimum production rates. Less grinding or burnishing media and less finishing, polishing or colouring compounds are required for a specific quantity of work. Visual observation of the work in progress is possible throughout the operation.

Machines can perform cleaning, descaling, deburring, radiusing, fine finishing, colouring, and burnishing operations. Extreme examples of their range of capabilities are severe cut-down of 0·025 in stock removal from stainless steel forgings and castings and delicate deburring and

Vibratory finishing machine
(Hepburn Conveyor Co. Ltd.)



Internal grinding attachment for Werner spline-shaft grinders
(Rockwell Machine Tool Co. Ltd.)

finishing without distortion of laminations 4 in diameter × 0·006 in thick. Currently, machines are being manufactured in capacities of $1\frac{1}{2}$, 3, 6, 12, and 18 ft^3 and the Company will undertake the layout and building of complete processing plants with all necessary equipment, and supply compounds and media as required.

Internal spline grinding

For the Type 3·275 range of four Werner automatic spline-shaft grinding machines, accepting shafts of from $\frac{3}{4}$ in to $3\frac{1}{2}$ in diameter in lengths up to 71 in, the manufacturers, Fritz Werner AG, West Berlin, have recently introduced an internal grinding attachment. Providing the number of splines is even, it enables the flanks of spline bores to be ground. Grinding spindles are available to cover bores from $\frac{13}{16}$ in to $3\frac{3}{8}$ in diameter. After the normal grinding spindle and the wheel dressing unit are removed, the attachment is mounted in the bore of the headstock. The grinding wheel is belt driven from a motor mounted vertically above the main housing. Cams for controlling the hydraulically operated cross movement of the internal grinding spindle are mounted on the machine table, as also is the special wheel-dressing device.

The workpiece is held in a three-jaw chuck mounted on the standard workhead, and a fine adjustment is provided for accurate positioning of the spline grooves. In operation, the lower flank of the rearmost spline groove is ground during the left-to-right travel of the hydraulic table. On reaching the reversing point the grinding spindle is automatically traversed hydraulically to the front so that during the reverse stroke of the table the lower flank of the diametrically opposite spline groove is ground.

On completion of the double stroke, when the table has reached its extreme left-hand position and the grinding wheel is clear of the work, the dividing head indexes automatically and brings the next groove into the horizontal position. The grinding spindle is also automatically traversed to its rear position and the grinding process repeated. By following this procedure, one grinding pass is carried out on all flanks as the workpiece is rotated through 360 deg. After resetting the grinding spindle in the vertical position the next grinding pass is undertaken on all flanks. This operating cycle is repeated until the finish dimensions of the spline grooves are reached.

Rockwell Machine Tool Co. Ltd., Welsh Harp, Edgware Road, London, N.W.2, are the sole selling agents for this and other Fritz Werner machines in the United Kingdom.

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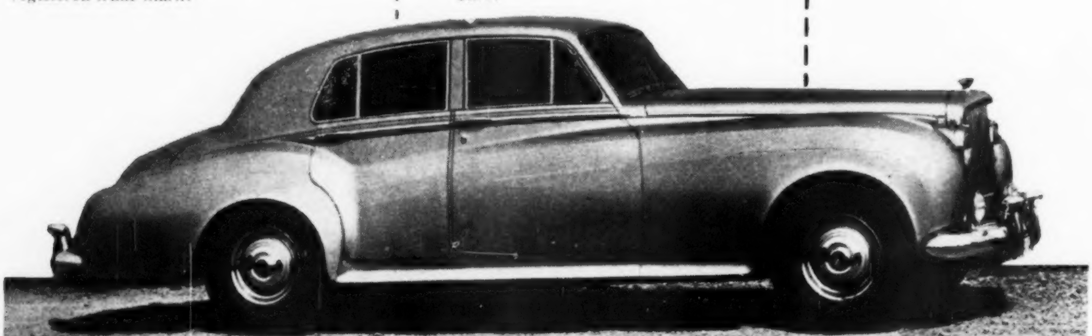

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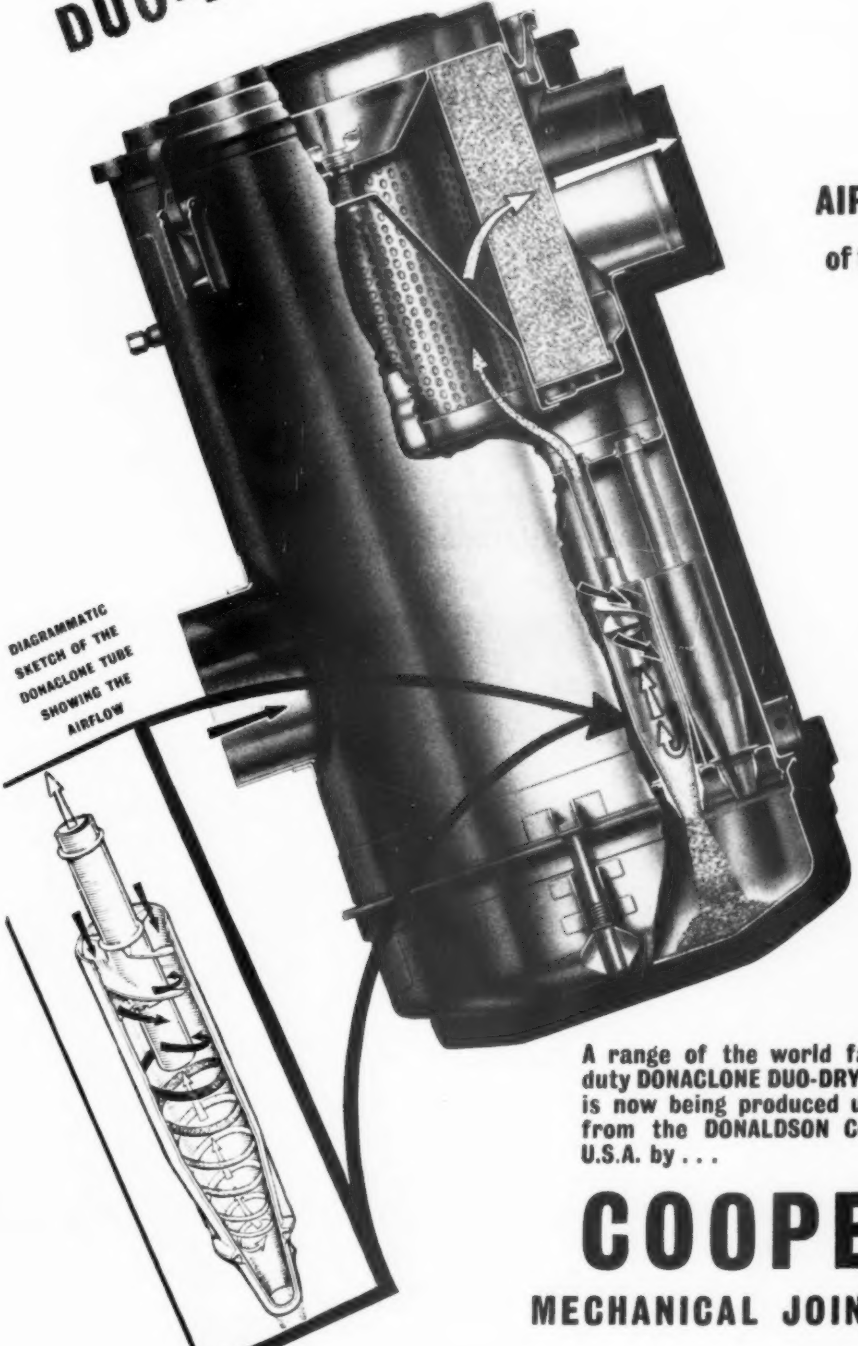

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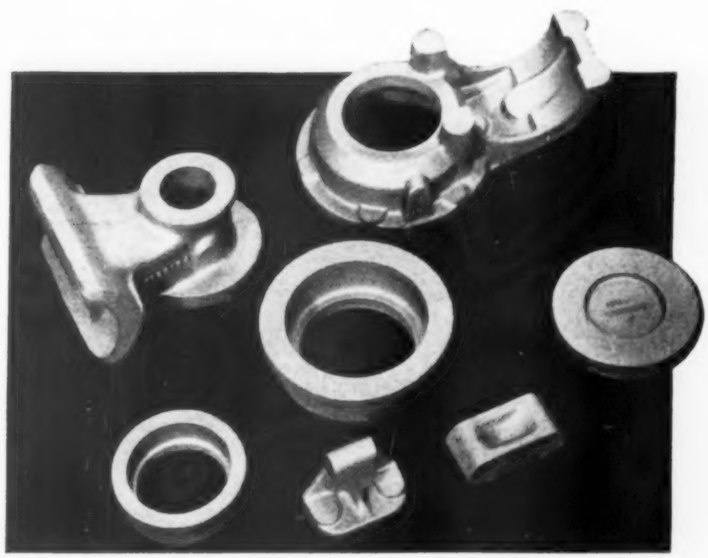

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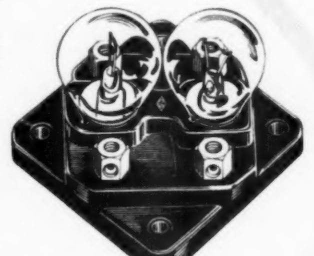

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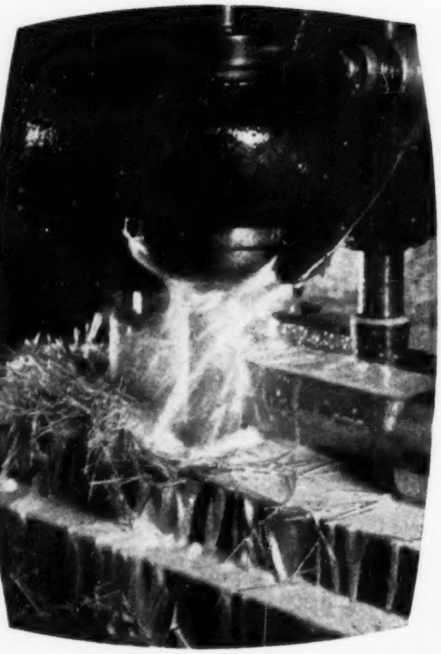

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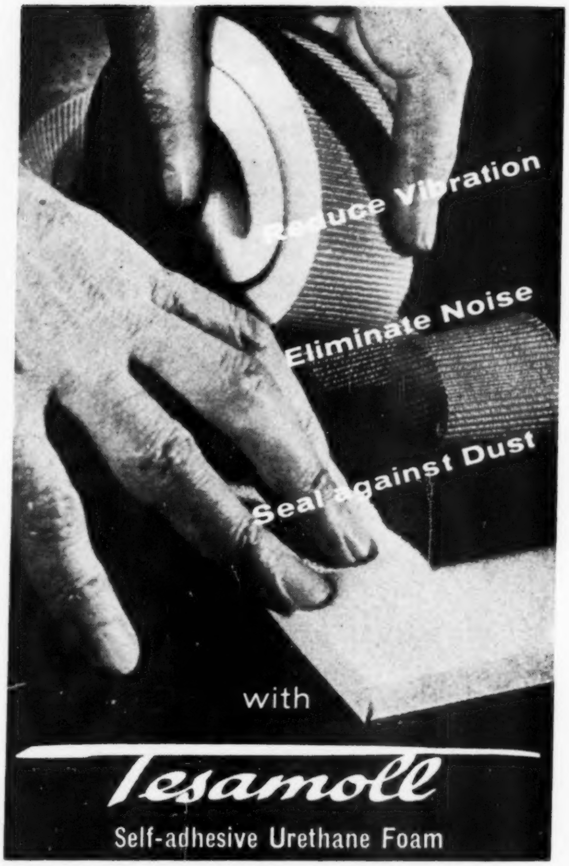

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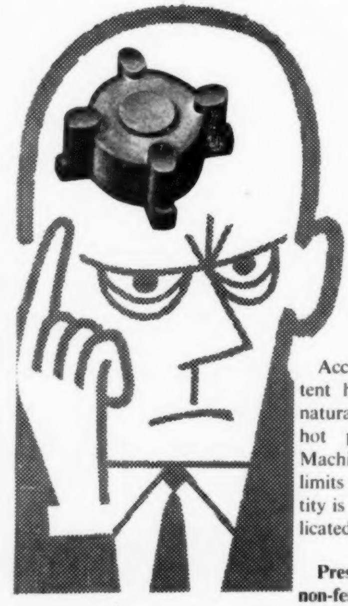

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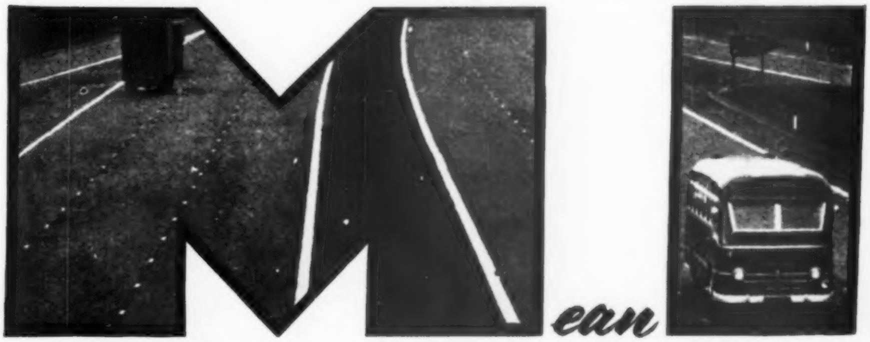

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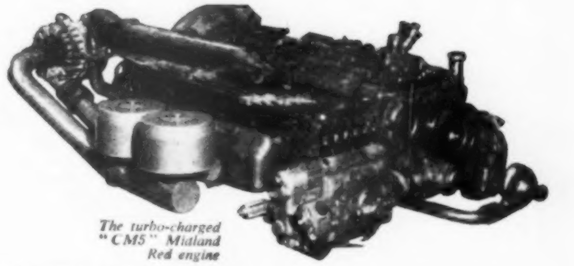

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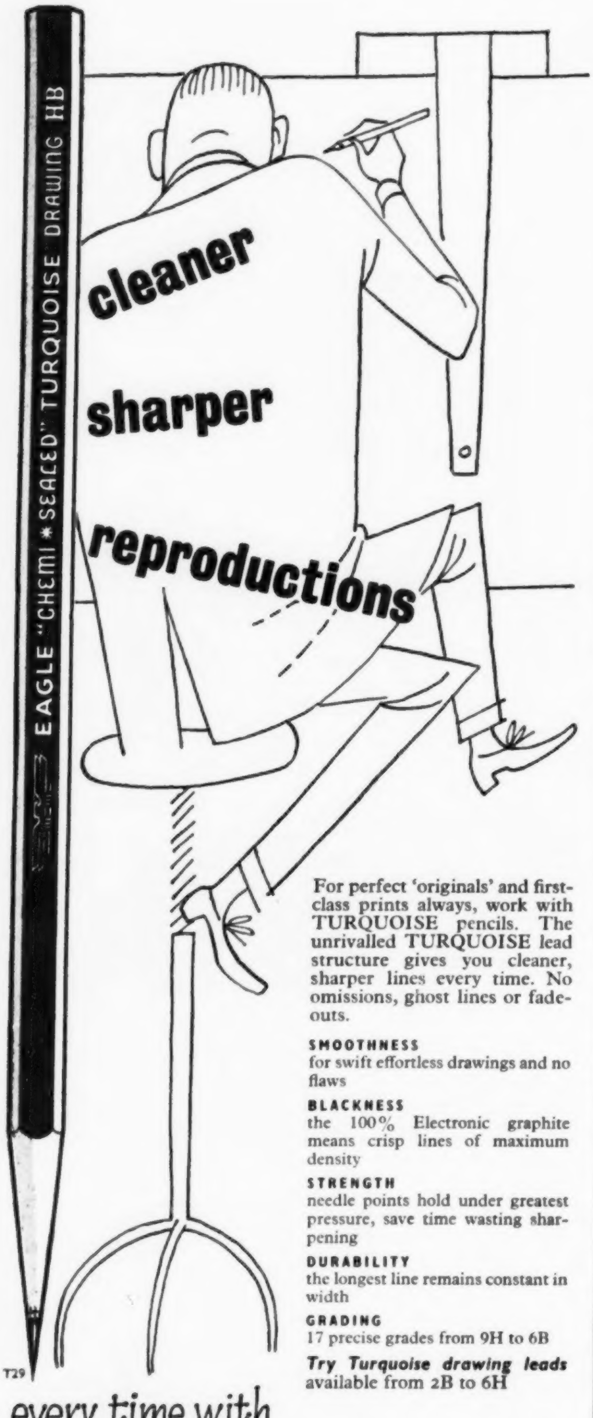

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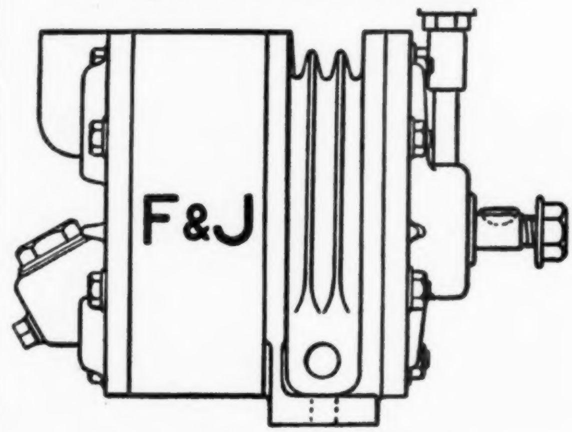

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